ANALYSIS OF COOLING CAPACITY AND OPTIMIZATION OF COMPRESSOR OUTLET PRESSURE FOR kW CLASS HELIUM REFRIGERATOR/LIQUEFIER

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

Master of Technology

In

Mechanical Engineering

By

Nishigandha Jadhav (212ME5326)

Department of Mechanical Engineering

National Institute of Technology

Rourkela

2014

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Under the Guidance of

Mr. A. K. Sahu.

(Institute for Plasma Research, Bhat, Gandhinagar),

Prof. Ranjit Kumar Sahoo.

Department of Mechanical Engineering

National Institute of Technology

Rourkela

2014

National Institute of Technology

Rourkela

CERTIFICATE

This is to certify that the thesis entitled, **"Analysis of Cooling Capacity and Optimization of Compressor Outlet Pressure for kW Class Helium Refrigerator / Liquefier"** submitted to **National Institute of Technology, Rourkela** by **Nishigandha Jadhav**, Roll No. **212ME5326 f**or the award of the Degree of **Master of Technology** in **Mechanical Engineering** with specialization in" **Cryogenic and Vacuum Technology**" is a record of bonafide research work carried out by her under my supervision and guidance. The results presented in this thesis have not been, to the best of my knowledge, submitted to any other University/ Institute for the award of any degree or diploma.

The thesis, in my opinion, has reached the standards fulfilling the requirement for the award of degree of **Master of Technology** in accordance with regulations of the Institute.

Mr. A. K. Sahu.
Scientist / Engineer – SF, Division Head,
Large Cryogenic Plant and Cryosystem,
Institute for Plasma Research,
Gandhinagar, Gujarat

Prof. R. K. SAHOO
Professor,
Department of Mechanical Engineering,
National Institute of Technology,
Rourkela.

CERTIFICATE

This is to certify that the dissertation, entitled

"Analysis of Cooling Capacity and Optimization of Compressor Outlet Pressure for kW Class helium Refrigerator / Liquefier"

Is a bonafide work done by

Nishigandha Jadhav

Under my close guidance and supervision in the Large Cryogenic Plant and Cryosystem Group

Of

Institute for Plasma Research, Gandhinagar, Gujarat

For the partial fulfillment of the award of the Degree of **Master of Technology** in **Mechanical Engineering** with specialization in "**Cryogenic and vacuum Technology**"

at

National Institute of Technology, Rourkela.

The work presented here, to the best of my knowledge, has not been submitted to any university For the award of similar degree.

GUIDE:

Mr. A. K. Sahu

Scientist / Engineer – SF, Division Head

Large Cryogenic Plant and Cryosystem,

Institute for Plasma Research,

Gandhinagar, Gujarat

ACKNOWLEDGEMENT

I avail this opportunity to express my indebtedness to my guide **Mr.A.K.Sahu**, Divisional Head of large cryogenics plant and Cryo system, Institute for Plasma Research, Bhat, Gandhinagar, Gujarat, and co-guide **Prof. R.K.Sahoo**, Mechanical Engineering Department, National Institute of Technology, Rourkela, for their valuable guidance, constant encouragement and kind help at various stages for the execution of this dissertation work. I consider myself fortunate to have worked under their supervision. I also express my sincere gratitude to **Dr. Sarangi**, Director of NIT Rourkela. **Prof. Maithy**, Head of The Department of mechanical Engineering at NIT Rourkela for providing valuable department facilities.

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Submitted By:
Nishigandha Jadhav
Roll No: 212ME5326
Department of Mechanical Engineering
National Institute Of Technology, Rourkela
Rourkela-769008.

Abstract

The main components of a helium liquefier which determines the performance of the HRL for a given compressor flow rate are Turbine, Heat exchanger and JT valve. Turbine and JT valve produces cooling effect of helium gas by isentropic and isenthalpic expansion process respectively. Different arrangement of components can be made to have different thermodynamic cycle configuration. For each configuration main components can have different operating process parameters leading to different performance of HRL. This project involves the analysis and optimization of compressor outlet pressure parameter for a given configuration. Normally JT valve is kept at the lowest temperature to get the highest liquefaction and this lowest temperature depends on the performance of other components and hence optimization of process parameter of JT valve is not considered here. One of the different cycle configurations is analyzed here and this cycle is often used in helium refrigeration and liquefaction plant. This configuration, planned to use for the indigenous helium plant, has three turbines and eight heat exchangers which produces liquid helium at 4.5 K. 1st and 2nd turbines operates at warmer temperature compared to 3rd turbine and s process flow paths of these warmer turbines are connected in series. Helium stream coming out of the 1st turbine passes to a heat exchanger which will reduce its temperature before entering the 2nd turbine. The nominal helium mass flow rate supplied by the compressor system is 210 g/s at pressure of 14 bars and 310 k temperature considering available standard compressors and capacity (~2 kW at 4.5 K) of the indigenous plant. Effects of compressor mass flow rate and pressure on the cooling capacity of the plant have been analyzed in this project. A part of this mass flow rate passes through a 1st and 2nd turbine for isentropic expansion to 1.2 bar and then this low pressure helium stream comes back to compressor suction through different heat exchangers to transfer cooling effect to the hot stream coming from the compressor. 3rd turbine will expand the remaining part of the main helium stream to 4 bar and this stream further passes through a heat exchanger before entering the JT valve for liquid helium production. This analysis and optimization work involves different practical factors and in efficiencies of main components. The analysis result for compressor delivery flow 140 g/s at 14 bar pressure is further compared with the performance of existing helium plant at IPR which has same compressor flow parameter. The results are also compared with that of the aspen tech software.

Keywords: Liquefaction, Helium, process parameters, Optimization, Turbine, heat exchanger.

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Chapter 1

INTRODUCTION

1.1 LIQUEFACTION OF GAS

Liquefaction is nothing but physical conversion of gas into liquid state and used for analyzing the fundamental properties of gas molecules, for storage of gases and in refrigeration and air conditioning. Many gases can be converted into gaseous state by simple cooling at normal atmospheric pressure and some other requires pressurization like carbon dioxide. Liquefaction is the process of cooling or refrigerating a gas to a temperature below its critical temperature so that liquid can be formed at some suitable pressure which is below the critical pressure. Using an ambient-temperature compressor, the gas is first compressed to an elevated pressure. This high-pressure gas is then passed through a counter-current heat exchanger to a throttling valve (Joule-Thompson valve) or an expansion engine. Upon expanding to a certain lower pressure below the critical pressure, cooling takes place and some fraction of gas is liquefied. The cool, low-pressure gas returns to the compressor inlet through a recycle stream to repeat the cycle. The counter-current heat exchanger warms the low-pressure gas prior to recompression, and simultaneously cools the high-pressure gas to the lowest temperature possible prior to expansion.

1.2 HELIUM

Air contains 78% of Nitrogen and 0.0005% of helium whereas both are used as cryogenic refrigerants. Thermodynamic properties of helium are given below:

Property Data / Fluid	4He
Normal boiling point (K)	4.22
Critical temperature (K)	5.20
Critical pressure (Bar)	2.3

Table1.2.1: Thermodynamic properties of Helium

The critical temperature of the fluid refers to the temperature of the critical point where the saturated liquid and saturated vapor states are identical. Liquefaction of helium (4He) with the Hampson-Linde cycle led to a Nobel Prize for Heike Kamerlingh Onnes in 1913. At ambient pressure the boiling point of liquefied helium is 4.22 K (-268.93°C). Below 2.17 K liquid 4He has many amazing properties, such as exhibiting super fluidity (under certain conditions it acts as if it had zero viscosity) and climbing the walls of the vessel. Liquid helium (4He) is used as a cryogenic refrigerant; it is produced commercially for use in superconducting magnets such as those used in superconducting Tokamak, MRI or NMR. Cryogenic technology is the study of production of very low temperature (below - 1500C or 123 K) and the behavior of materials at those temperatures. For the liquefaction process, development of such low temperature working device, air separation and fundamental principles and procedures have been discussed in well-known text books of cryogenics [1-5].This chapter discusses several of the systems used to liquefy the cryogenic fluids.

1.3 HELIUM LIQUEFIER/REFRIGERATOR

Helium liquefier as the name suggest is used for the liquefaction process of Helium gas. The cold box shown below is used for the cool down and liquefaction purpose of Helium gas coming out of the Tokomak. Cold box contains total 8 heat exchangers and 3 turbines which expand isentropic ally and one JT valve which expands isenthalpic ally. Process parameters of heat exchanger are effectiveness or UA, mass flow rate, Temperatures and for turbine are temperatures, mass flow rate, inlet outlet pressure; efficiency has to be optimized to get maximum liquefaction of LHe with minimum refrigeration load.

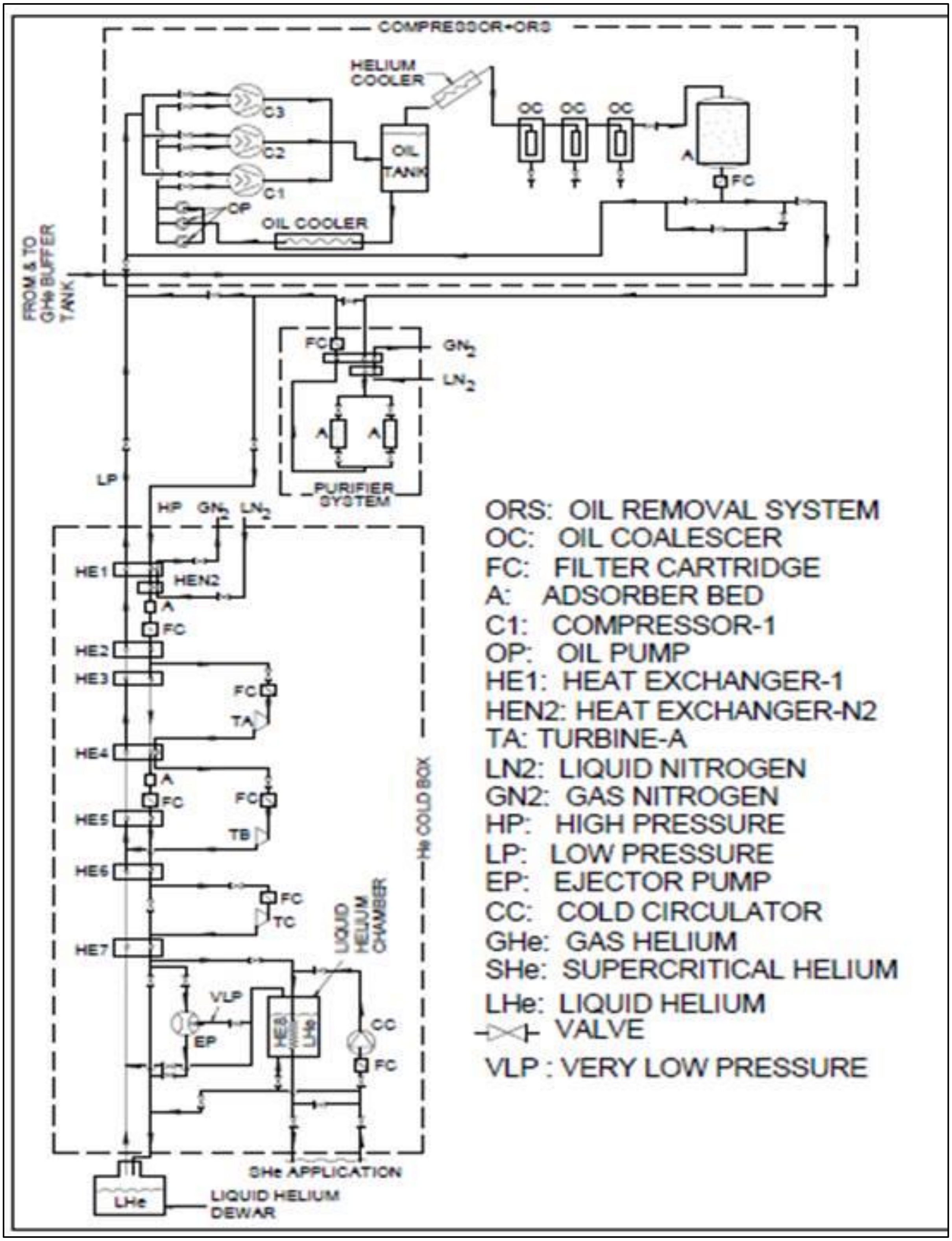


Figure 1.3.1: Typical Schematic of the cold box along with the warm and cold end components for Helium plant of Tokamak

1.4 THE THERMODYNAMICALLY IDEAL SYSTEM

Thermodynamically ideal liquefaction system is firstly used for the comparison of liquefaction systems through the figure of merit. This system is ideal in the thermodynamic sense, but it is not ideal as far as practical system is concerned. Carnot cycle is the perfect cycle in which ideal liquefaction is having a reversible isothermal compression followed by a reversible isentropic expansion. The gas to be liquefied is compressed reversibly and isothermally from ambient conditions to some high pressure. This high pressure is selected so that gas will become saturated liquid upon reversible isentropic expansion through the expander. The final pressure is taken as the same as the initial pressure. The pressure attained at the end of isothermal compression is extremely high in the order of 70 GPa and it is not an ideal process for a practicable system as it is impracticable to handle such a pressure. [1].

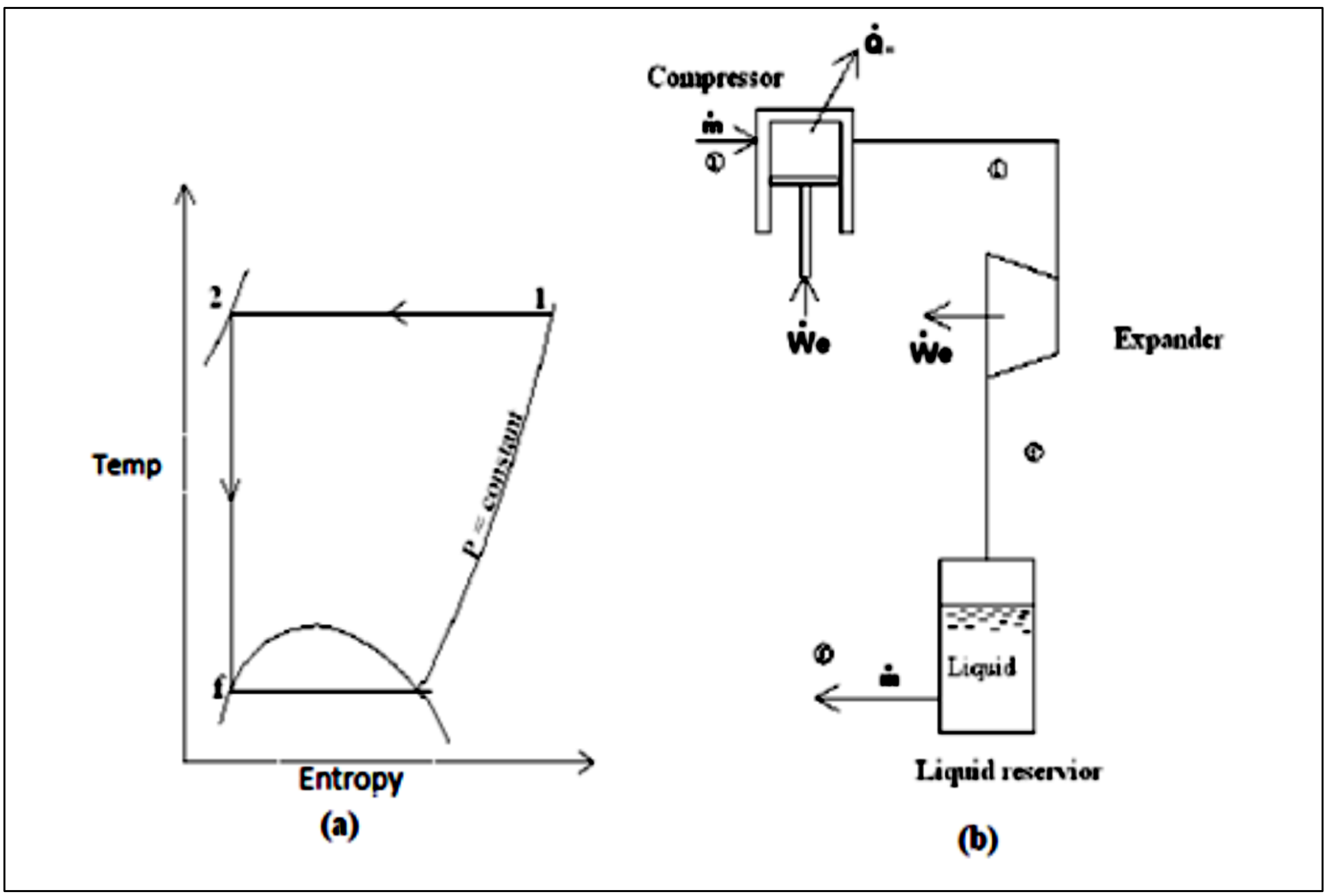


Figure1.4.1: (a) Thermodynamic cycle T-S plane (b) Apparatus Set-up [1].

The First law of thermodynamic for steady flow may be written as:

$$Q_{net} – W_{net} = \text{Outlet } mh - \text{Inlet } mh$$

Applying the First law to the system shown in figure:

$$QR – W1 = m (hf – h1)$$

The heat transfer process is reversible and isothermal in the Carnot cycle. Thus, from the second law of Thermodynamics:

$$QR = mT1 (S2 – S1) = - mT1 (S1 – Sf)$$

Because the process from point 2 to point f is isentropic, S2 = S3, where S is the entropy of the fluid. Substituting QR, we may determine the work requirement for the ideal system:

$$- (Wi/m) = T1 (S1 – Sf) – (h1 – hf)$$

1.5 PRODUCTION OF LOW TEMPERATURE

1.5.1 Joule–Thompson effect

Most of the practical liquefaction systems produce low temperatures using either an expansion valve or a Joule Thomson valve. Applying the first law for steady flow to the expansion valve, for zero heat and work transfer and for negligible kinetic and potential changes, we find h1= h2. Flow within the valve is irreversible and is not an isenthalpic process; the inlet and the outlet do lie on the same enthalpy curve. We note that there is a region in which an expansion through the valve produces an increase in temperature; while in another region the expansion results in a decrease in temperature. Obviously we should operate the expansion valve in a liquefaction system in the region where there is a net decrease in temperature results. The curve that separates two regions is called the inversion curve. The effect of change in temperature for an isenthalpic change in pressure is represented by the Joule-Thompson coefficient [1].

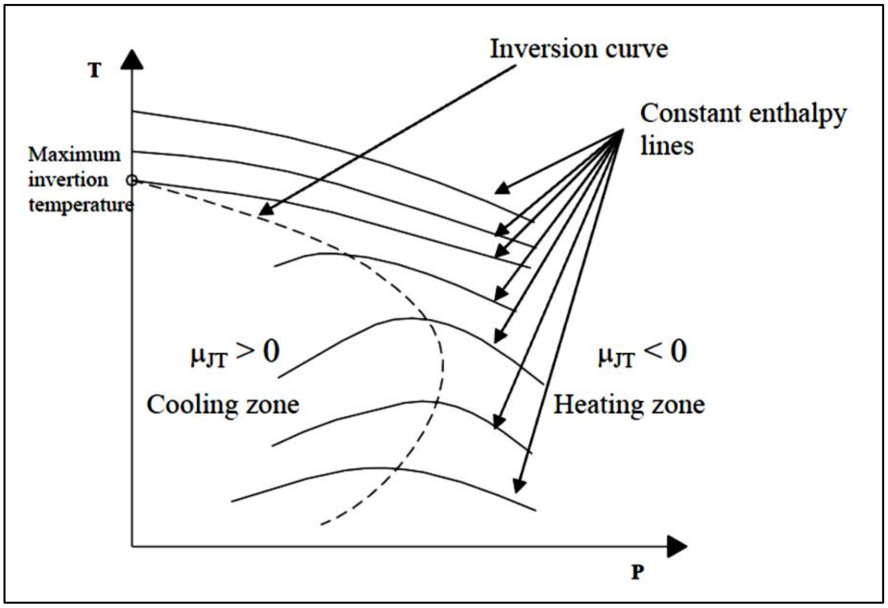


Figure1.5.1.1: Isenthalpic expansion of a real gas [1].

1.5.2 Adiabatic expansion

The second method of producing low temperatures is the adiabatic expansion of the gas through a work producing device, such as an expansion engine. In the ideal case, the expansion would be reversible and adiabatic and therefore isentropic. In this case we can define the isentropic coefficient which expresses the temperature change due to a pressure change at constant entropy [1]. Isentropic outlet temperature (T6s) is calculated for the turbine from isentropic relation for an ideal gas is $T6s = T2*(P6/P2)^{(r-1)/r}$ and then actual temperature (T6a) is found out from the turbine isentropic efficiency using formula: $\eta = (T2-T6a) / (T2-T6s)$

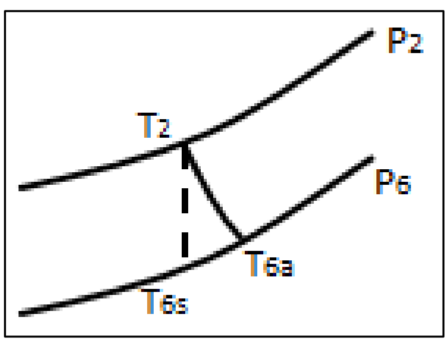


Figure 1.5.2.1: Isentropic expansion of a Turbine

1.6 THERMODYNAMIC CONFIGURATION FOR HELIUM PLANT

1.6.1 Collins helium liquefaction system:

The Collins cycle or the modified Claude cycle is the one which is normally used for helium liquefaction. Figure 1.6.1.1 (a) gives a schematic diagram of the Collins cycle and (b) gives its process representation on the T-S diagram. HX1, HX2… HX6 are the nomenclature for the six heat exchangers used in this liquefaction system and EX1 and EX2 are the two reciprocating expanders as shown in the schematic diagram below. m is the total mass flow rate of the helium gas through the compressor while me1 and me2 are the mass flow rates diverted through the expansion engine number 1 and 2, respectively. mf is the liquefaction yield. Ph and Pl represent discharge and suction pressure of the compressor [6].

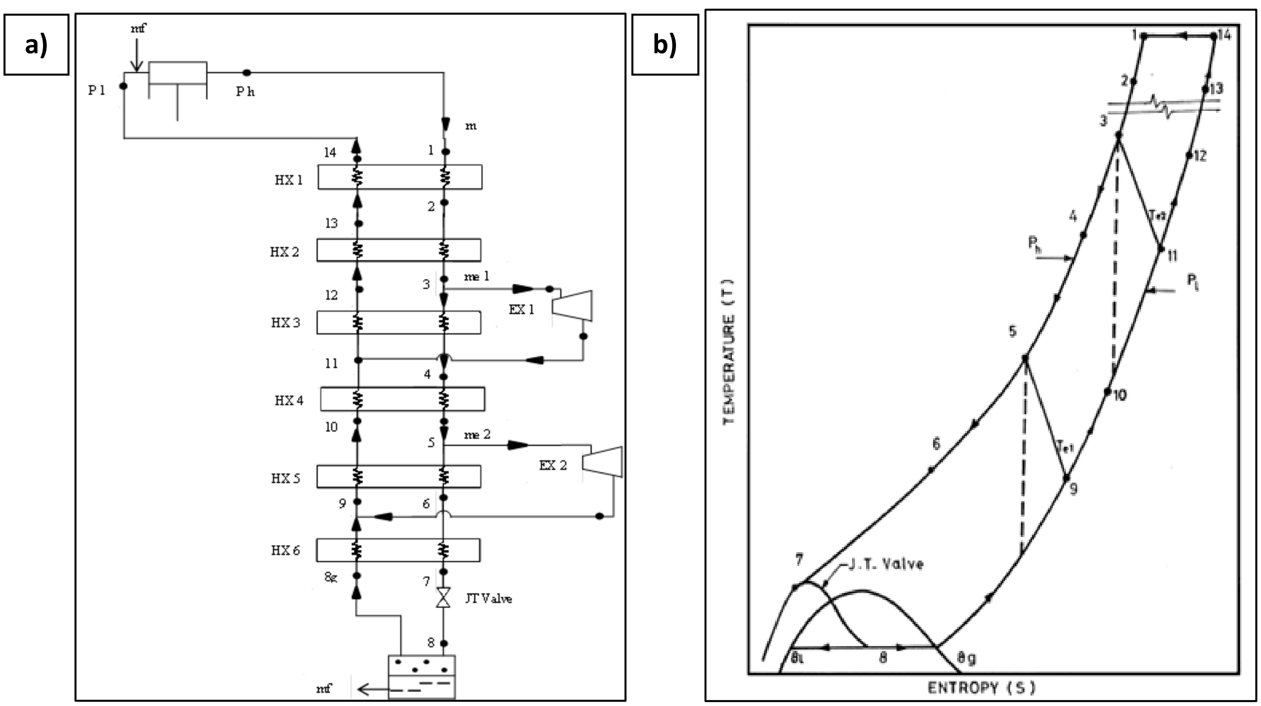


Figure1.6.1.1: Collins Helium (a) Liquefaction Cycle (b) T-S diagram

1.6.2 Assumptions in Collins Helium Liquefaction system:

- The maximum pressure (Ph) in the system is 15 bar and the minimum pressure (Pl) is 1 bar.
- The temperature of the gas after compression is 300 K which the ambient temperature and the return stream temperature of the helium gas after liquefaction is at its boiling point, i.e. 4.21 K.
- The pressure drop in the heat exchangers is negligible.
- The J-T expansion is a perfect isenthalpic expansion process.
- Heat in-leak in the system is negligible.
- Effectiveness of heat exchangers and efficiencies of expanders are assumed to be constant and their dependence on pressure, temperature and mass flow rate is ignored.

1.6.3 Analysis and Performance of the system:

The thermo physical properties of the helium gas, at different temperatures and pressures, are taken from Van Sciver [6]. For any intermediate temperatures, the values for enthalpy, entropy, etc. are linearly interpolated. Applying the first law of thermodynamics to the system, excepting the compressor, for the steady state condition, the ratio of liquid yield to the total mass flow rate, y, is given as follows: where, x1 = me1 / m and x2 = me2 / m. del he1 and del he2 are the net enthalpy changes in helium occurring in EX1 and EX2 respectively. h represents enthalpy at the respective points. Different parameters like heat exchanger effectiveness (ε), expander efficiencies (n1 and n2), temperatures of gas before expansion, total mass flow rate (*m*), mass flow fraction through expanders (*me*1 + *me*2) etc. affect the performance of the liquefier. The cold produced in the expanders is directly proportional to the mass flow rate diverted through them and the liquefaction yield is proportional to the remaining mass flow rate that passes through the J-T valve.

Chapter 2

LITERATURE REVIEW

M.D. Atrey [7], suggest the effect of expander efficiency and heat exchanger effectiveness on the performance of the liquefier in a Collins helium liquefaction cycle with two reciprocating expanders. It states that for a given efficiency of expanders and effectiveness of heat exchangers, there exists an optimum mass flow fraction of total helium gas mass flow rate that should be diverted through the expanders for which liquid yield is maximum and net power input is minimum. It gives final steady state temperature distribution across the cycle, which is necessary for carrying out the preliminary design of various components in the cycle.

G. Cammarata., A. Fichera., D. Guglielmino, [8] gave an optimization methodology for liquefaction/refrigeration systems in the cryogenic field with Figure of Merit as the evaluation index, and genetic algorithms as evaluation criteria. This methodology has been applied to an existing helium liquefaction system that works according to a Collins cycle which allows optimizing the system by taking suitable number of independent variables that are sufficient to characterize the plant. Optimized the liquid helium production in considered application with maximum mass flow rate conditions, gives an improvement of 10% in the FOM.

D. Henry, J.Y. Journeaux, P. Roussel, F. Michel, J.M. Poncet, A. Girard, V. Kalinin, P. Chesny [9], states that CEA is carrying out an analysis of the various ITER cryoplant operational modes. ITER has designed to be operated 365 days per year to optimize the available time of the Tokamak. Operation is running for a long time but separated by a maintenance period with annual or bi-annual major shutdown periods of a few months. Auxiliary subsystems like the cryoplant and the cryodistribution have to cope with different heat loads which depend on the different ITER operating states. Cryoplant consists of four identical 4.5K refrigerators and two 80K helium loops coupled with two LN2 modules. All these systems are operated in parallel to remove the heat loads from the magnet, cryopumps and other small users. A new design consists of updated layout of the cryodistribution system, refrigeration loop for the HTS current leads and revised strategy for operations of the cryopumps. Plasma operation state, short term stand by, short term maintenance, or test and conditioning state are normal operating scenarios of the cryoplant which are checked for the typical ITER operating states. Last part of the paper presents the abnormal operating modes of the magnets and generated by the cryoplant. The occurrence of a fast discharge or a quench of the magnets generates large heat loads disturbances and produces exceptional high mass flow rates which have to be managed by the cryoplant, while a failure of a cryogenic component induces a major disturbance for the magnet system. Because of this analysis modifications are made in the present PFD to match the technical specifications of the cryogenic system with the ITER operation requirements.

R. J. Thomas, P. Ghosh, K. Chowdhury [10], stated that efficiency of helium liquefiers used in fusion reactors can be calculated by the performance of their constituting components like heat exchanger. On simulating with Aspen HYSYS V7.0, the effects of heat exchanger process parameters in a helium liquefier can be understood. Independent parameter UA (product of overall heat transfer coefficient U, heat transfer surface area A and deterioration factor F) which takes into account all thermal irreversibility and configuration effects. Rate of liquefaction is directly proportional to UA, saturates at limiting UA and shows the linear variation with the effectiveness of heat exchangers. Also has influence on performance of those heat exchangers that determine the inlet temperatures to expanders. Variation of sizes of heat exchangers does not affect the optimum mass flow rate through expanders. When effectiveness remains equal for all heat exchangers gives the maximum liquefaction.

R. J. Thomas, P. Ghosh, K. Chowdhury [11], performed the parametric studies using Aspen HYSYS® 7.0 and results are extrapolated to understand the behavior of large scale helium liquefiers. It shows that the maximum liquid production is obtained when 80% of the compressor flow is diverted through the expanders and it is equally distributed between the two expanders in a Collins cycle analysis. Liquid production and the isentropic efficiency of expanders show the linear relationship which is same for the higher and lower temperature expanders.

Rijo Jacob Thomas, Parthasarathi Ghosh, Kanchan Chowdhury [12], suggest that in a helium liquefiers/ refrigeration expanders connected in parallel (reverse Brayton stage) or in series and also in series with heat exchangers between them (modified Brayton stages). Using exergy analysis the options of splitting and combining Brayton stages into modified Brayton stages are evaluated. Results show that the performance is not good when two Brayton stages are combined to make two modified Brayton stages. When one Brayton stage is split into two modified Brayton stages, the performance shows improvement with the total heat exchanger surface area remaining unchanged. Splitting led to more improvement when the stage operates at lower temperatures. Each stages either Brayton or modified Brayton has been found to behave as independent refrigeration stage allowing more additions of heat exchanger area. At any temperature of operation, brayton stage has been found to be superior to a modified Brayton stage while doing one to one comparison. When heat exchangers in the configuration are less balanced in mass flow the impact of replacing Brayton stage with modified Brayton stage has been found to be more pronounced.

Partho S. Roy and Ruhul Amin M. [13], states increasing demand of gas production against low production rate at the time of energy crisis effects the domestic and industrial operations as natural gas is major power source. There is a dwindling situation in gas production as almost all plants are operating beyond limits. Establishment of a new gas plant and other power sources has made the situation complicated. In such a case optimization of the gas plant is the only better way. This paper presents the steady state simulation of Bakhrabad gas processing plant (at Sylhet) using the Aspen HYSYS shall be performed based on both the design and physical property data of the plant.

Rijo Jacob Thomas, Parthasarathi Ghosh, Kanchan Chowdhury [14], Suggest that exergy is a useful tool for analyzing and optimizing the design and operation of systems. There are some literature serves available on helium refrigerators and liquefiers based on exergy. This paper evaluates the operating and geometric parameters to determine the exergy destructions in components as well as in the entire cycle of Collins helium liquefiers. Grassmann diagram of exergy flow helps in understanding relative importance of different components used in the system. Compressor pressure, expander flow rates, heat exchanger surface area are some of the parameters optimized considering both presence and absence of pressure drop in the heat exchangers. For a plant of any capacity results are applicable using Non-dimensionalization of parameters. Exergy-analysis based on Second Law is far superior to the First Law based energy analysis in designing of the helium plant and capable of deriving some additional conclusions. Derived results from a Collins cycle may be applicable in large-scale helium liquefiers by giving basic knowledge of the components on the plant performance and reasonable initial guess values in their design and simulation.

QIU Lilong., ZHUANG Ming., MAO Jin., HU Liangbing., SHENG Linhai [15], Suggest a designed steady-state program to simulate the primary cycle of the EAST cryogenic system and compressor is tested to obtain the best isothermal efficiency. The actual *UA* values of the heat exchanger, turbine efficiencies and fraction of the mass flow rate have been analyzed to optimize the cycle. An equivalent refrigeration capacity is used to evaluate the refrigerator in different operation modes. Finally, an upgraded mode is proposed on the basis of the calculations and estimate of future heat loads from the tokamak device.

Rijo Jacob Thomas, Rohan Dutta , Parthasarathi Ghosh, Kanchan Chowdhury[16], gives the proper design of helium systems with large number of components and involved configurations such as helium liquefiers/refrigerators requires the use of tools like process simulators. Simulations results are accurate as per the accuracy of given data. The 32-parameter MBWR equation of state proposed by McCarty and Arp [19] for computation of thermodynamic properties of helium is widely used. It is computationally involved which makes the simulation process more time-consuming and sometimes leads to computational difficulties such as numerical oscillations, divergence in solution especially, when the process operates over a wide thermodynamic region and is constituted of many components. Substituting MBWR EOS by simpler equations of state (EOS(s)) at selected thermodynamic planes, where the simpler EOS(s) have the similar accuracy as that of MBWR EOS may enhance ease of computation. This paper has been adopted with the above mentioned methodology with an example of steady as well as dynamic simulation of helium liquefier/refrigerator based on Collins cycle. The above concept can be applied to the computation of fluid property which involves the thermodynamic analysis of other process cycles.

Chapter 3

METHODOLOGY

3.1 DIFFERENT METHODS TO ANALYZE THE PROCESS PARAMETERS OF HEAT EXCHANGERS AND TURBINE:

Indigenous helium plant of about 2 KW cooling capacity is planned to be built at Institute for Plasma Research (IPR). Existing helium plant has about 1.3 kW cooling capacity and similar arrangement of heat exchangers and turbines. The planned indigenous helium plant's process flow diagram and T-S diagram are shown in Fig 3.1.1 before going to do thermodynamic analysis of such bigger system, simpler systems have been tried to develop certain analysis methods. These methods are further analyzed to produce a hybrid method as per the requirement for the convergence of all temperatures of heat exchangers. Then it is easy to optimize using that method on various process parameters such as Effectiveness, UA, mass flow rate for heat exchanger and pressure, mass flow rate, efficiency for a turbine.

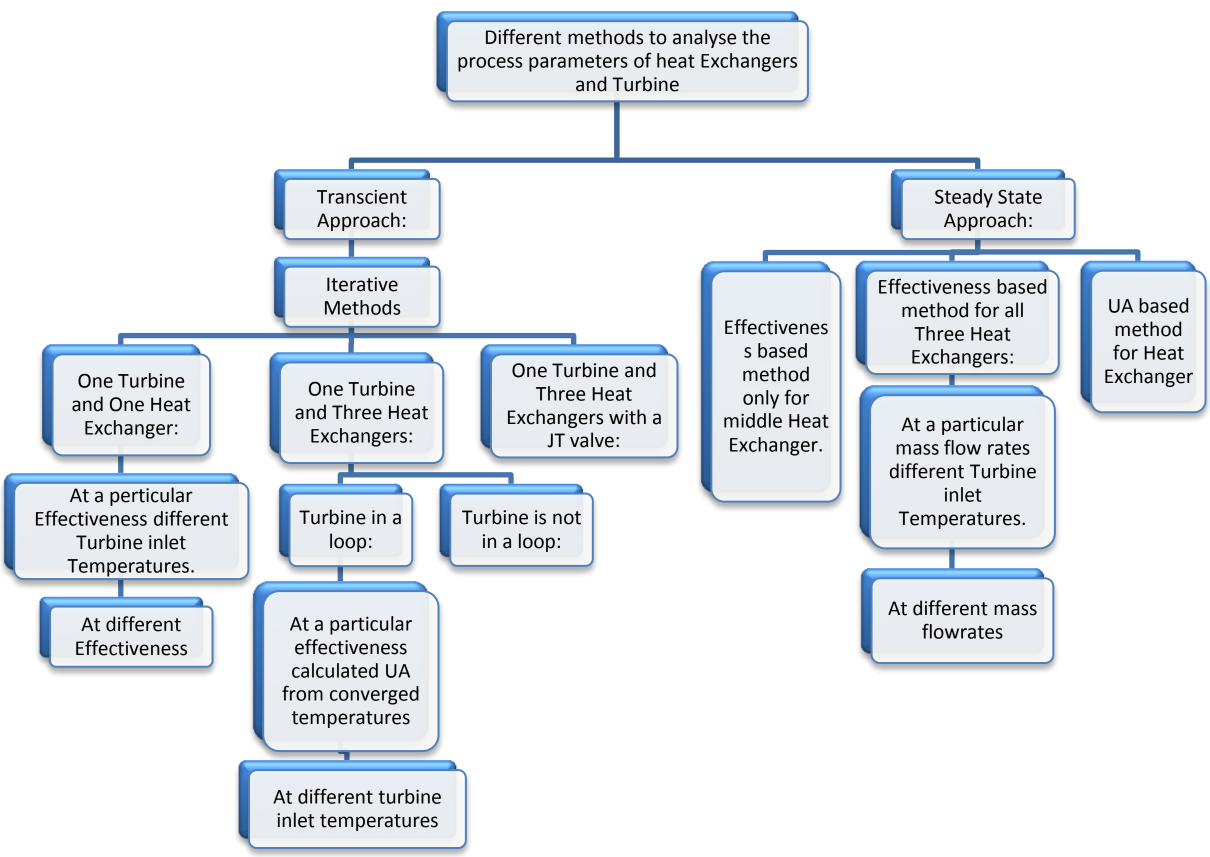


Figure 3.1.1: Different methods to analyze the process parameters of Heat Exchangers and Turbine.

3.2 TRANSIANT APPROACH:

3.2.1 ONE TURBINE AND ONE HEAT EXCHANGER: (HELIUM)

1) **At a particular Effectiveness different Turbine inlet Temperatures**
   a) **At different Effectiveness:**

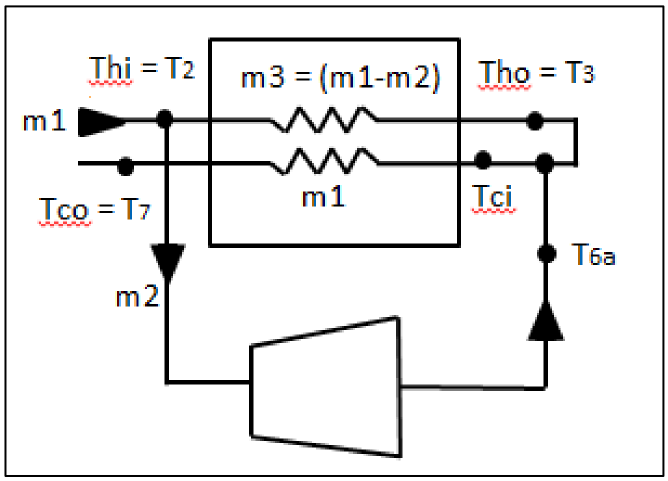
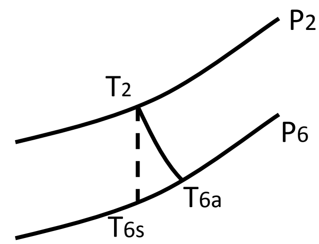


Figure 3.2.1.1: One Turbine and One Heat Exchanger

- Let turbine inlet temperature (Thi) =300K fixed for all the iterations and Effectiveness of a heat exchanger (E) =0.9.Turbine expands isentropic ally from 14 bar (P2) to 1 bar (P6) with a turbine isentropic efficiency = 70%
- Total mass flow rate through a compressor (m1) =100g/s of which 50% is diverted to the turbine (m2) and rest (m3) passes through the hot stream of the heat exchanger. At the inlet of the cold stream of heat exchanger, mass flow rates add up and gives the total mass flow rate same as through the compressor (m1).
- Jot down the valves of Cp from the fluid database software (NIST). Cph at 14 bar pressure, 300 K temperature and Cpc at 1 bar pressure, 150 K temperature.
- Isentropic outlet temperature (T6s) is calculated for the turbine from isentropic relation for an ideal gas is

$$T6s = T2*(P6/P2)^{(r-1)/r}$$



- Then actual temperature (T6a) is found out from the turbine isentropic efficiency using formula:

$$\eta = (T2-T6a) / (T2-T6s)$$

- For the initial case let Tho = Thi = 300K and inlet cold stream temperature of a heat exchanger can be calculated using formula:

$$T6 = ((T6a*m2) + (Tho*m3)) / m1$$

- Assign heat exchanger cold stream temperature (Tci) = T6.
- Calculate the heat exchanger hot stream outlet temperature (Tho) and cold stream outlet temperature (Tco) using effectiveness formula:

$$E = ((m*Cp)\ h *(Thi - Tho)) / ((m*Cp)\ min * (Thi-Tci))\ \text{OR}$$

$$E = ((m*Cp)\ c *(Tco - Tci)) / ((m*Cp)\ min * (Thi-Tci))$$

- Now, in the second iteration Thi = 300 K which is fixed and Tci= T6 can be calculated using Tho from previous iteration and T6a. Then with the use of effectiveness formula new Tho and Tco can be calculated. Again the same procedure repeats till Tho, Tci and Tco will get converged.

3.2.1.2 Flow chart for one turbine and one heat exchanger:

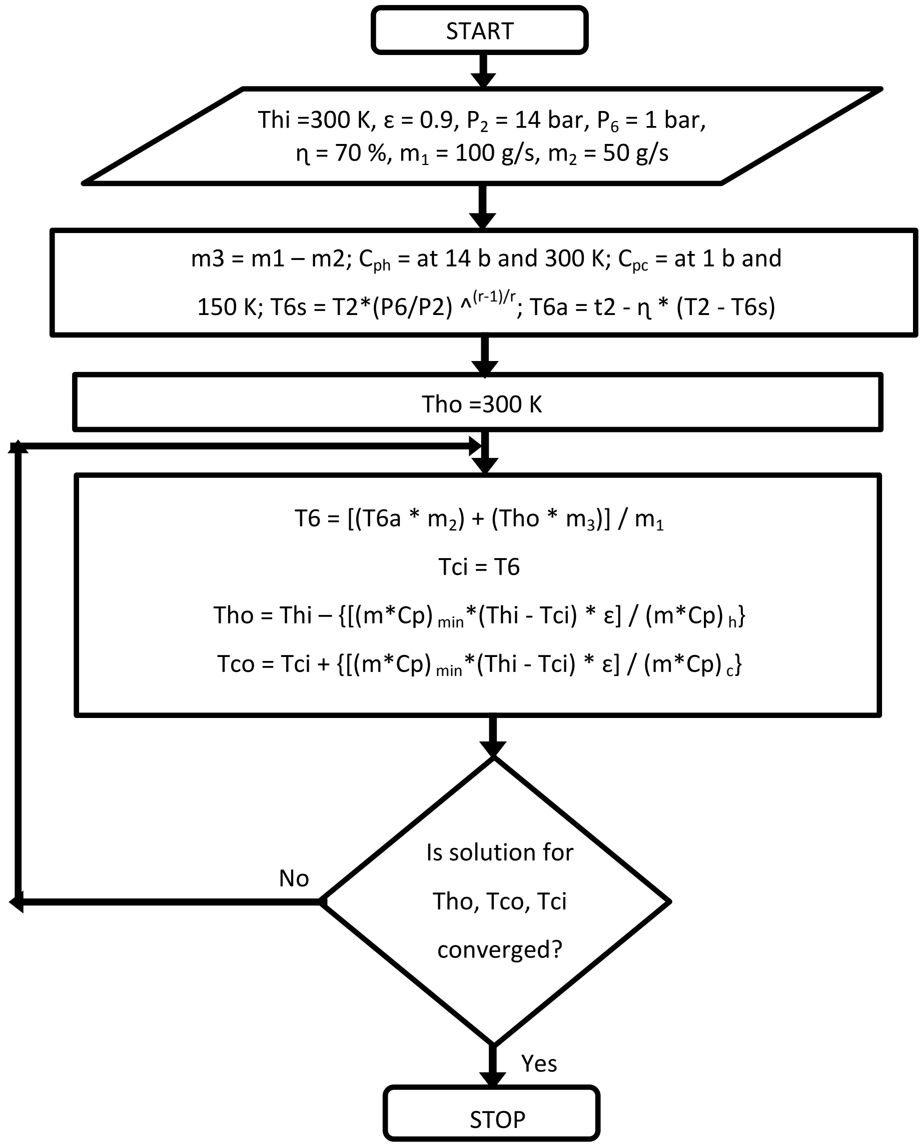


Figure 3.2.1.2: Flow chart for one turbine and one heat exchanger.

- Above method can be carried out for
  - ✓ heat exchanger effectiveness (E) = 0.9 at different fixed Turbine inlet temperatures (Thi) = 300K, 200K, 150K

- ✓ Heat exchanger effectiveness (E) = 0.8 at different fixed Turbine inlet temperatures (Thi) = 300K, 200K, 150K and results are plotted on a graph to see whether it has been converged or not.

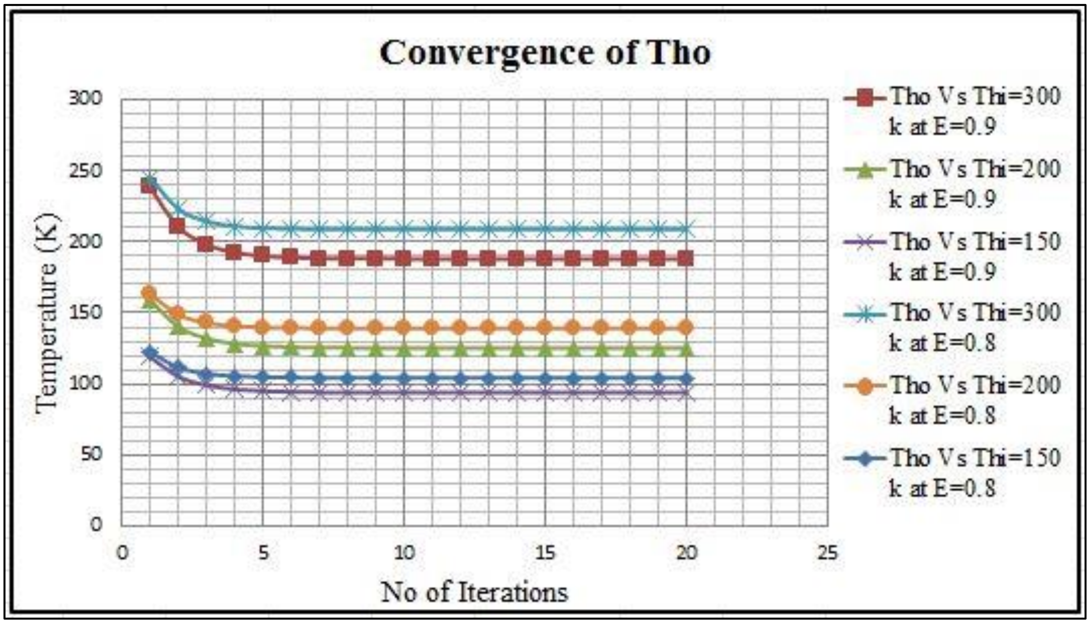


Figure 3.2.1.3: Plot of converged temperatures of one Turbine and One Heat Exchanger

- Conclusion from the above plots:
  - ✓ all the temperatures are converged with this method and
  - ✓ Higher value of effectiveness gives the better heat transfer.

3.2.2 ONE TURBINE AND THREE HEAT EXCHANGERS:

1. Turbine in a loop:

(a) At a particular effectiveness calculated UA from converged temperatures

(i) At different turbine inlet temperatures:

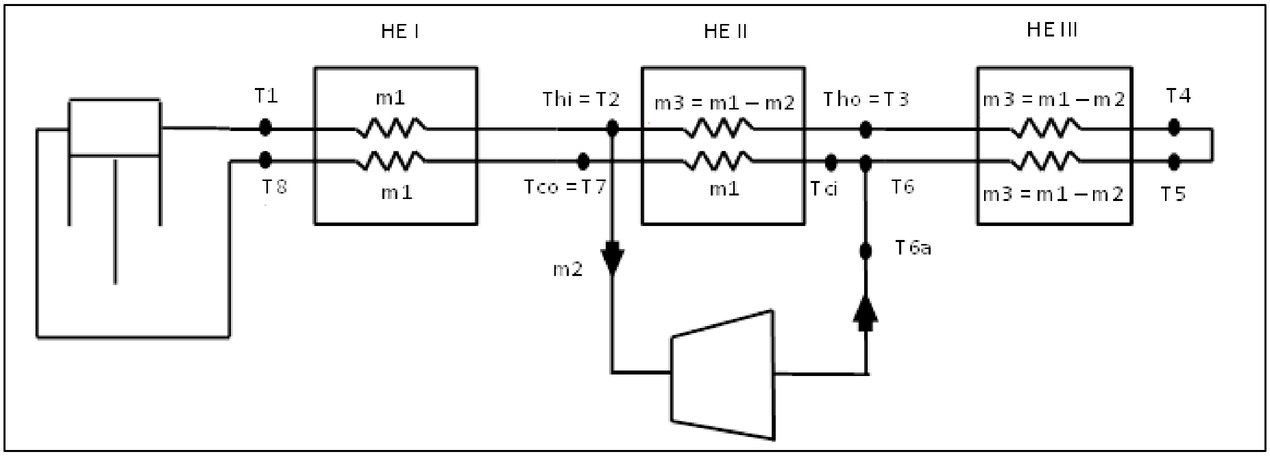
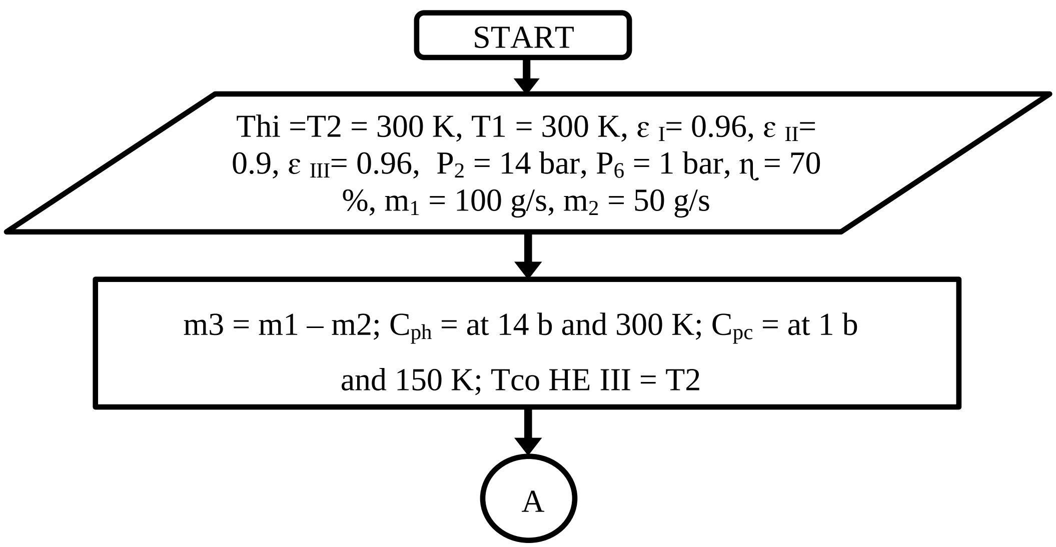


Figure 3.2.2.1: Schematic diagram of one turbine and one heat exchanger.

3.2.2.1 Flow chart for one turbine and three heat exchanger using effectiveness and turbine is in loop:



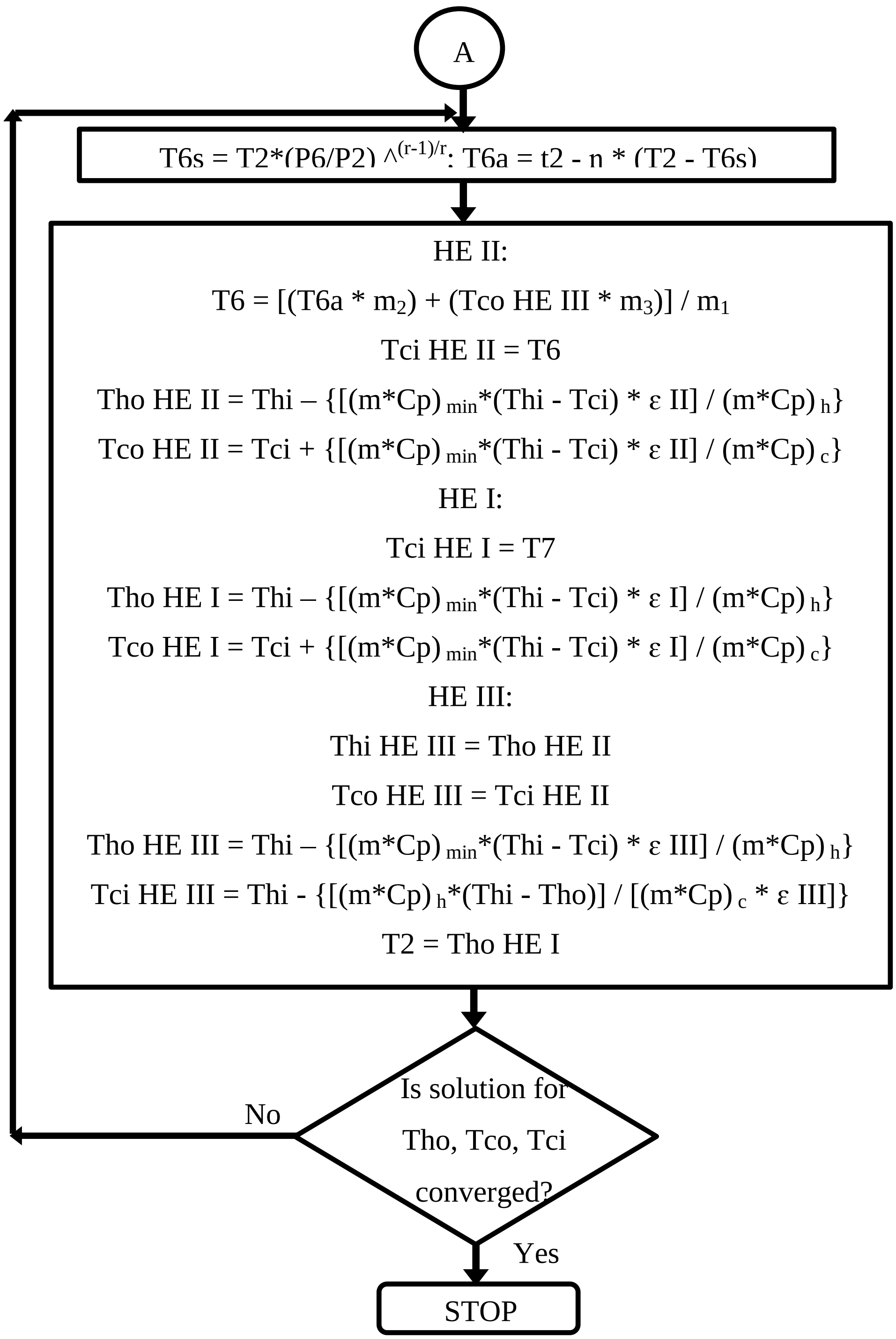


Figure 3.2.2.2: Flow chart for one turbine and one heat exchanger.

- Above method can be carried out for:
    - Heat exchanger (I, II, III) effectiveness (E) = 0.96, 0.9, and 0.96 respectively at different fixed Turbine inlet temperatures (Thi II) = 300K...

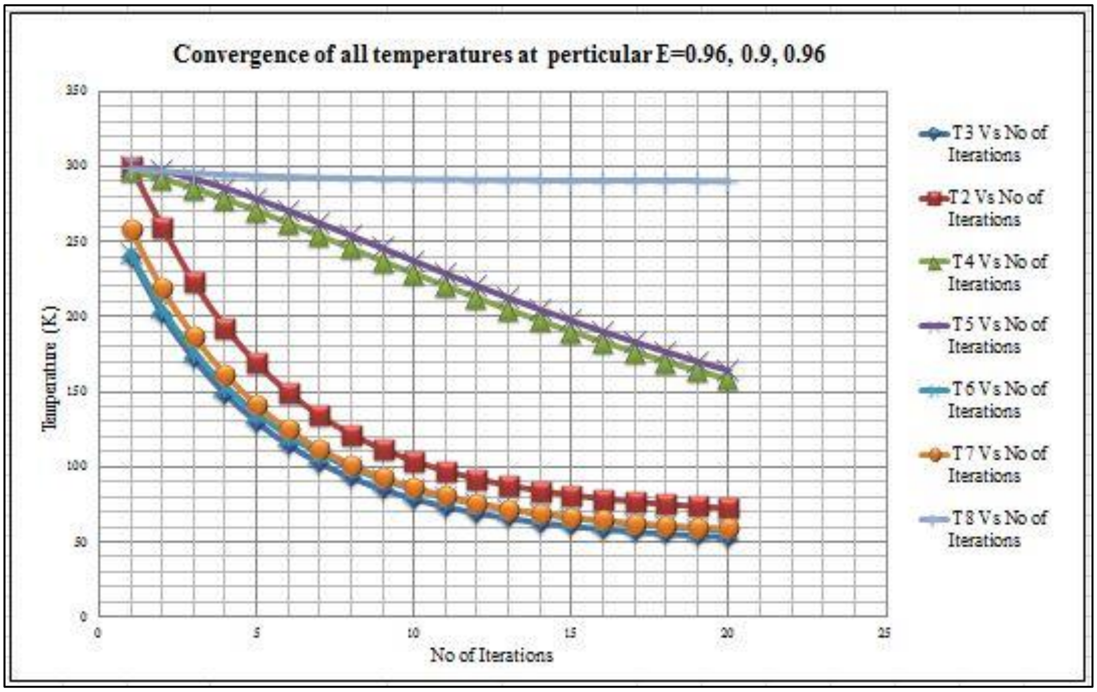
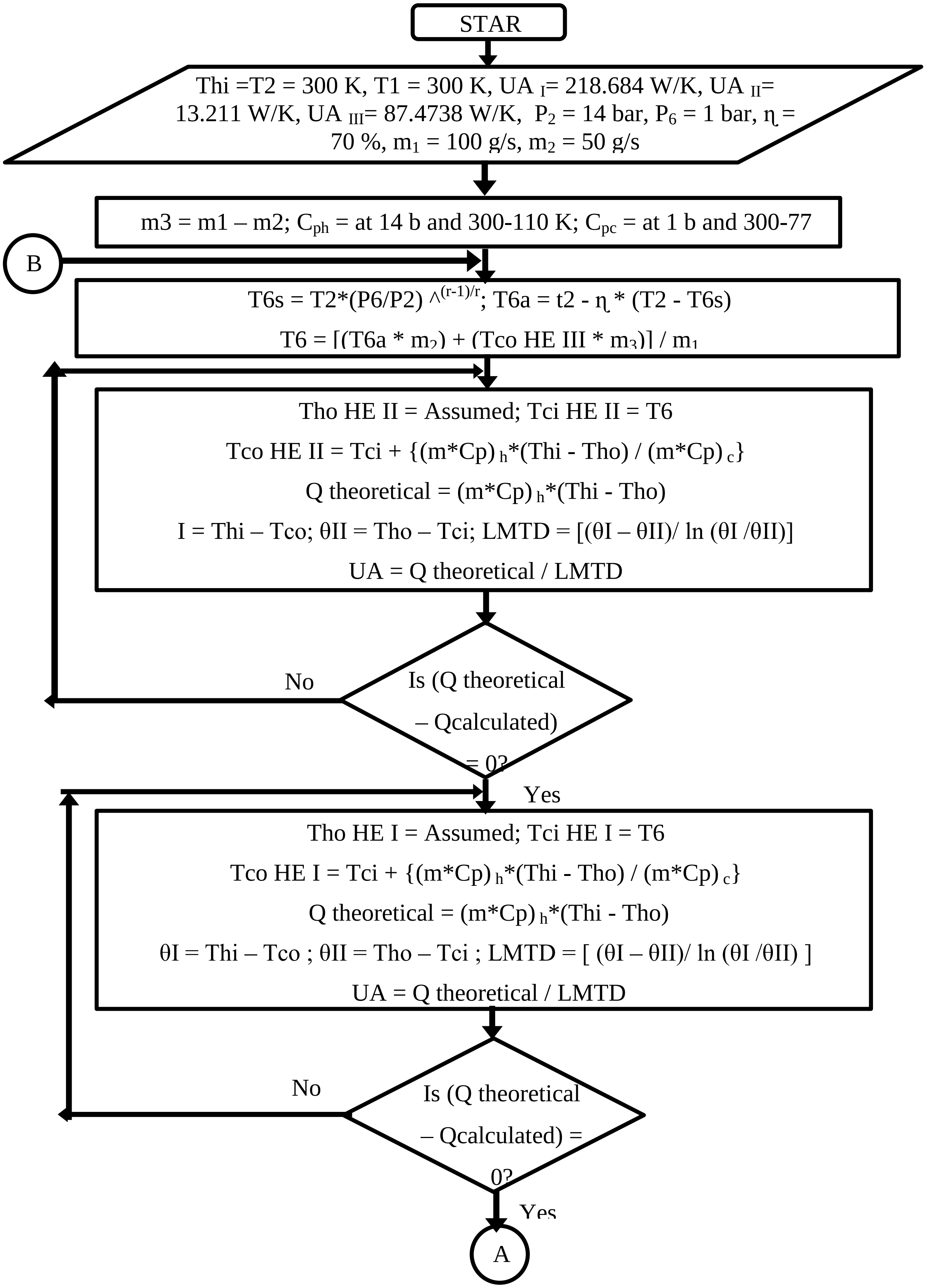


Figure 3.2.2.3: Plot of converged temperatures of one Turbine and Three Heat Exchanger using different effectiveness of all Heat Exchangers.

ii. Different valves of UA for Heat Exchanger:

- From this plot UA values can be calculated at the converged points and those are UA (I, II, III) values =218.684 W/K, 13.211 W/K, 87.4738 W/K respectively.

3.2.2.4 Flow chart for one turbine and three heat exchanger using UA and Turbine in loop:



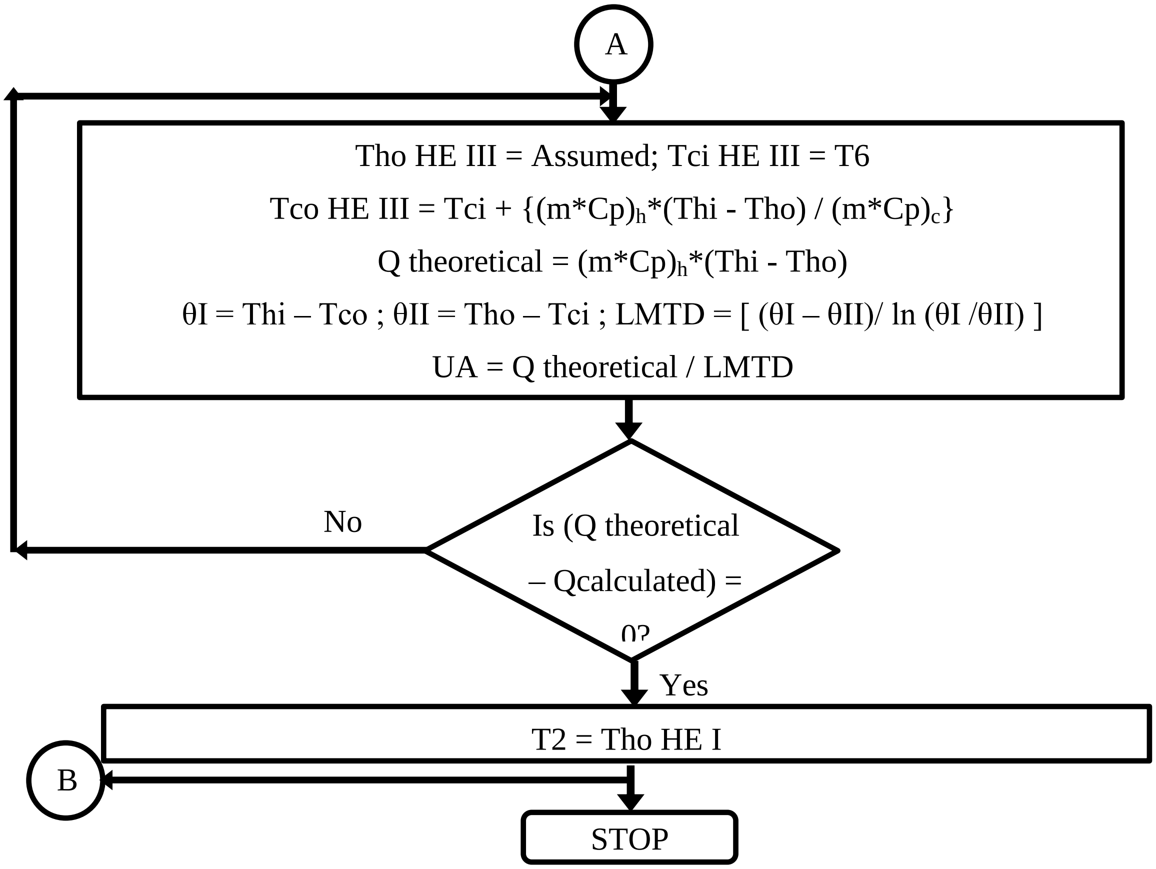


Figure 3.2.2.4: Flow chart for one Turbine and Three Heat Exchanger using calculated UA and turbine is in a loop.

- Same method follows till the temperatures will get converged.

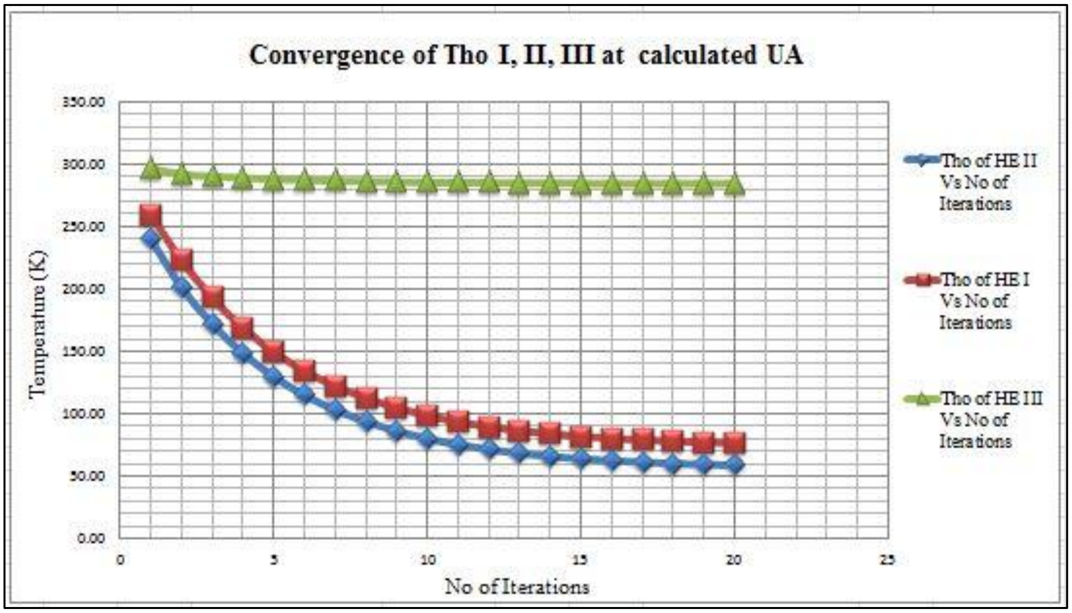


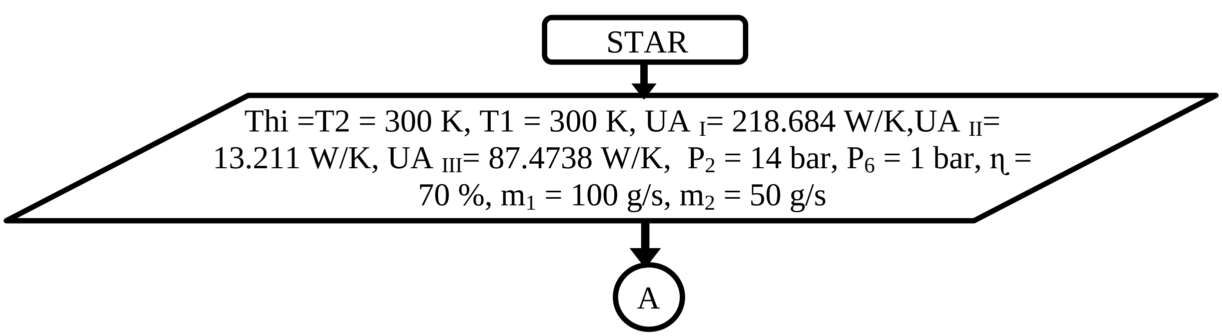
Figure 3.2.2.5: Plot of converged temperatures of one Turbine, Three Heat Exchanger using calculated UA and Turbine is in loop.

- Conclusion from the above plots:
  - ✓ All the temperatures are converged with this method. These methods can be used for the optimization.

1. Turbine is not in a loop:

- This method does not give a good cooling effect as turbine is expanding isentropic ally only once.
- Same procedure repeats for several times, till get the converged results.

3.2.2.6 Flow chart for one turbine and three heat exchanger using UA and Turbine is not in loop:



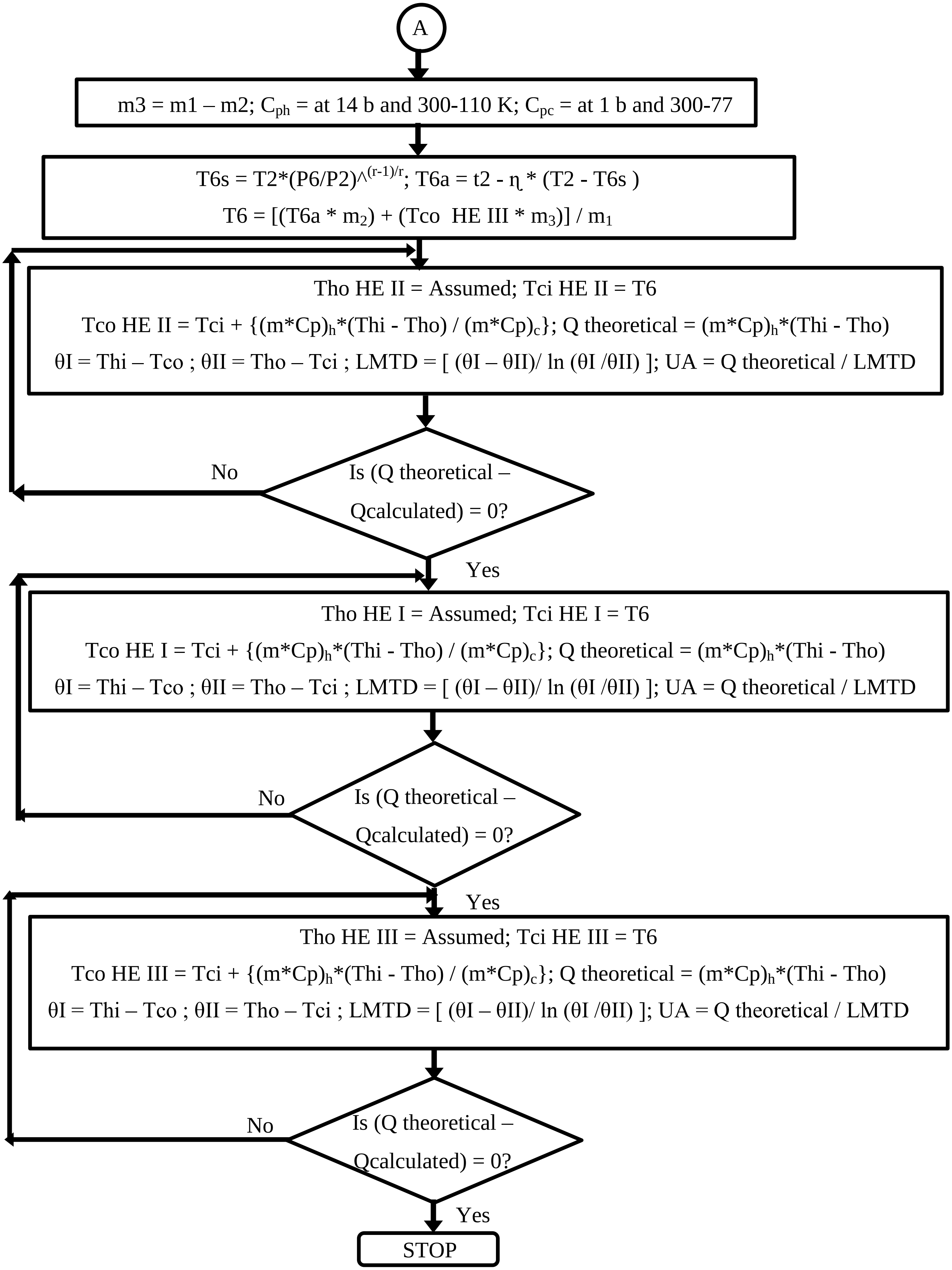

A
m3 = m1 – m2; C_{ph} = at 14 b and 300-110 K; C_{pc} = at 1 b and 300-77
T6s = T2*(P6/P2)^$^{(r-1)/r}$; T6a = t2 - ƞ * (T2 - T6s)
T6 = [(T6a * m_2) + (Tco HE III * m_3)] / m_1
Tho HE II = Assumed; Tci HE II = T6
Tco HE II = Tci + {$(m*Cp)_h$*(Thi - Tho) / $(m*Cp)_c$}; Q theoretical = $(m*Cp)_h$*(Thi - Tho)
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]; UA = Q theoretical / LMTD
No
Is (Q theoretical – Qcalculated) = 0?
Yes
Tho HE I = Assumed; Tci HE I = T6
Tco HE I = Tci + {$(m*Cp)_h$*(Thi - Tho) / $(m*Cp)_c$}; Q theoretical = $(m*Cp)_h$*(Thi - Tho)
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]; UA = Q theoretical / LMTD
No
Is (Q theoretical – Qcalculated) = 0?
Yes
Tho HE III = Assumed; Tci HE III = T6
Tco HE III = Tci + {$(m*Cp)_h$*(Thi - Tho) / $(m*Cp)_c$}; Q theoretical = $(m*Cp)_h$*(Thi - Tho)
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]; UA = Q theoretical / LMTD
No
Is (Q theoretical – Qcalculated) = 0?
Yes
STOP

Figure 3.2.2.6: Flow chart for one Turbine and Three Heat Exchanger using calculated UA and turbine is not in a loop.

3.2.3 ONE TURBINE AND THREE HEAT EXCHANGERS WITH A JT VALVE:

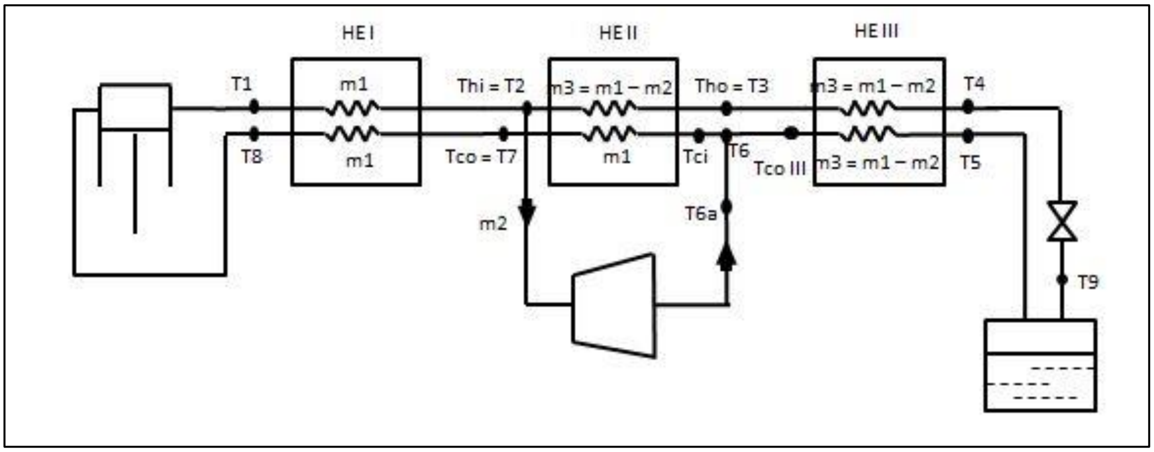


Figure 3.2.3.1: One Turbine and Three Heat Exchangers with a JT valve: Transient approach

- In this method exact same approach need to be used like one turbine with three heat exchangers but in addition to that some more points to be noted as follows:
  - ✓ Enthalpy can be calculated at 40 bar pressure and Tho III temperature.
  - ✓ As JT valve expands isenthalpic ally so h4 = h9, assuming some value of temperature at T9 and by using goal seek enthalpy difference (h4-h9) tends to zero gives the corrected T9 temperature.

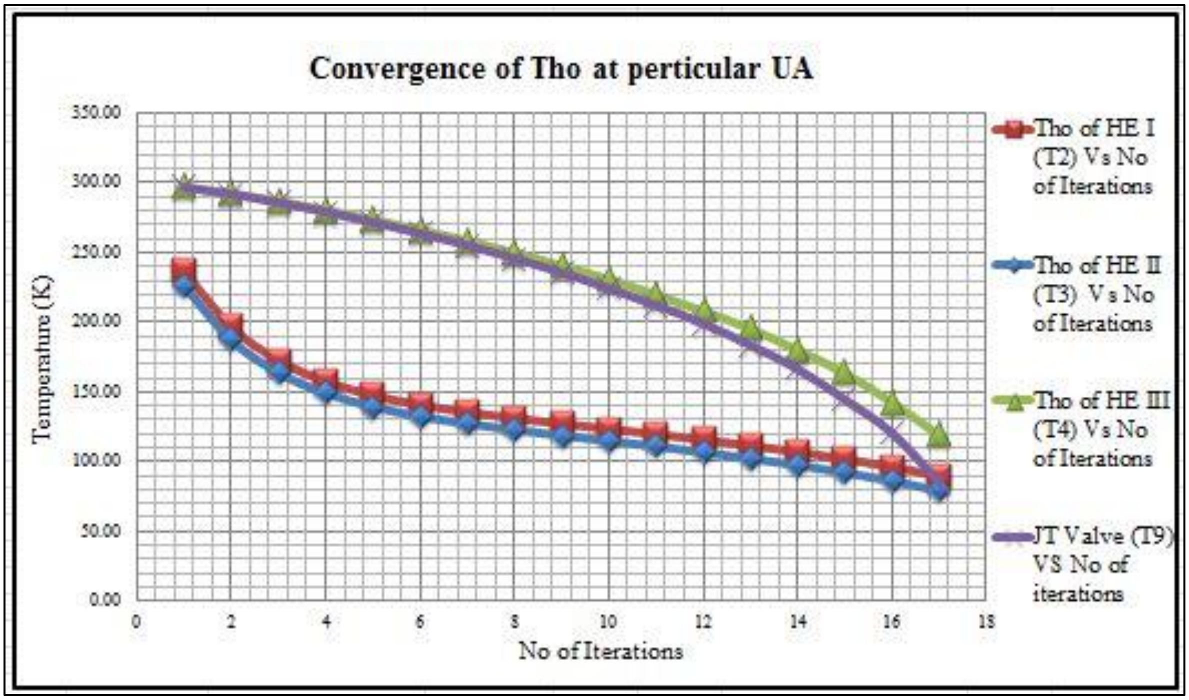


Figure 3.2.3.2: Plot of converged temperatures for one Turbine and Three Heat Exchangers with a JT valve: Transient approach

- Conclusion from the above plots:
  - ✓ All the temperatures for heat exchanger I and II are converged but the temperature for heat exchanger III and JT outlet are still decreasing and reaching to 84 K after 20th iteration. These methods can be used for the optimization.

3.3 STEADY STATE APPROACH:

3.3.1. EFFECTIVENESS BASED METHOD FOR ALL THREE HEAT EXCHANGERS: (Nitrogen)

(1) At a particular mass flow rates different Turbine inlet Temperatures:

(a) At different mass flow rates:

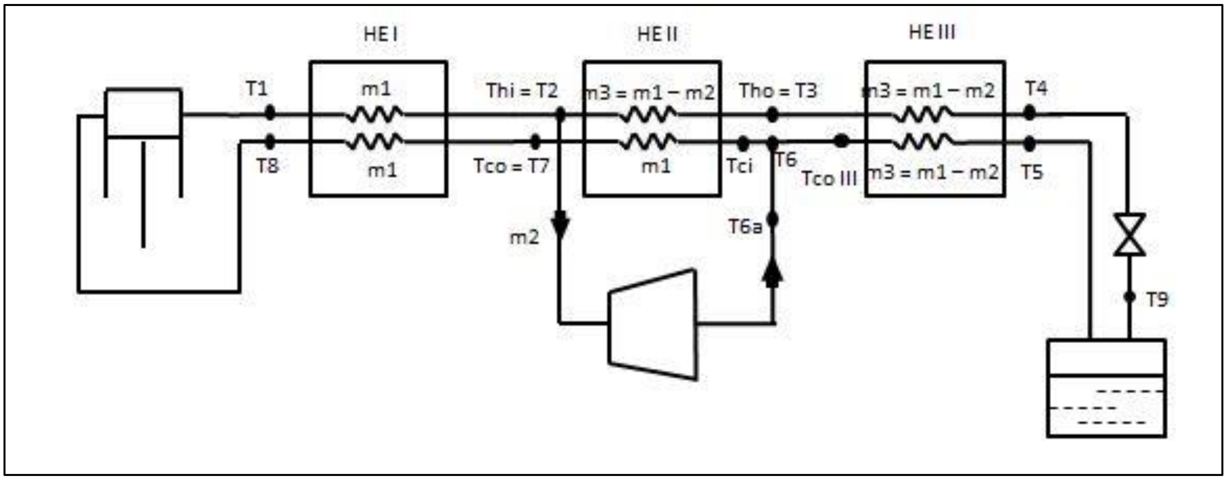


Figure 3.3.1.1: One Turbine and Three Heat Exchangers with a JT valve: Steady state approach

- This is a steady state approach in which effectiveness of each heat exchanger is 0.95, 0.9, and 0.9 respectively. Assuming different turbine inlet temperatures as 140 K, 160 K, 180 K, 200 K, 220 K, and 240 K at a fixed mass flow rate from a turbine (m2).
- Outlet temperature and pressure of a compressor is 300 K and 40 bar. Assuming inlet temperature of turbine is 140 K at 40 % mass flow rate from turbine (m2=0.04 kg/s).
- Thi I = 300 K, Tho I = 140 K, calculating Tci I and Tco I using effectiveness of heat exchanger I.

$$E = ((m*Cp)\ h *(Thi - Tho)) / ((m*Cp)\ min * (Thi-Tci))\ \text{OR}$$

$$E = ((m*Cp)\ c *(Tco - Tci)) / ((m*Cp)\ min * (Thi-Tci))$$

  - Turbine expands from 40 bar to 1 bar at inlet temperature of 140 K and found out actual turbine outlet temperature using isentropic expansion and adiabatic efficiency of a turbine.
- Isentropic outlet temperature (T6s) is calculated for the turbine from isentropic relation for an ideal gas is

$$T6s = T2*(P6/P2)^{(r-1)/r}$$

- And then actual temperature (T6a) is found out from the turbine isentropic efficiency using formula:

$$\eta = (T2-T6a) / (T2-T6s)$$

- Assigning Tho I = Thi II and Tci I = Tco II, assuming some arbitrary value to Tho II calculating Tci II using effectiveness formula.

$$E = ((m*Cp)\ h *(Thi – Tho)) / ((m*Cp)\ min * (Thi-Tci))$$

- Temperature difference (Tho II – Tci II) has to be managed using goal seek to get Tci II = T6a and corrected Tho II which was assumed.
- Calculated Tci III using energy balance,

$$(T6* m1)= (T6a*m2) + (Tco\ III *m3)$$

- Assigning Tho II = Thi III, assuming Tho III and knowing Tco III found out Tci III. Maintaining temperature difference (Tho III – Tci III) using goal seek so that Tci III should reach to a boiling point temperature of nitrogen at 1 bar which is 77 K and corrected Tho III.
- Calculating enthalpy at Tho III at 40 bar (h4) which will be same as h9 as helium gas expands isenthalpic ally using JT valve, calculating vapor and liquid fraction from h9 and refrigeration load and LN2 production as follows:

$$h4 = h9 = hf +x\ (hg - hf)$$

hf = Saturated liquid at 1 bar, hg = Saturated vapor at 1 bar, (hg-hf) = Latent heat, x = Vapor fraction, (1-x) = Liquid fraction.

$$\text{LN2 Production} = (1-x) * m3$$

$$\text{Refrigeration load} = (1-x) * hfg * m3$$

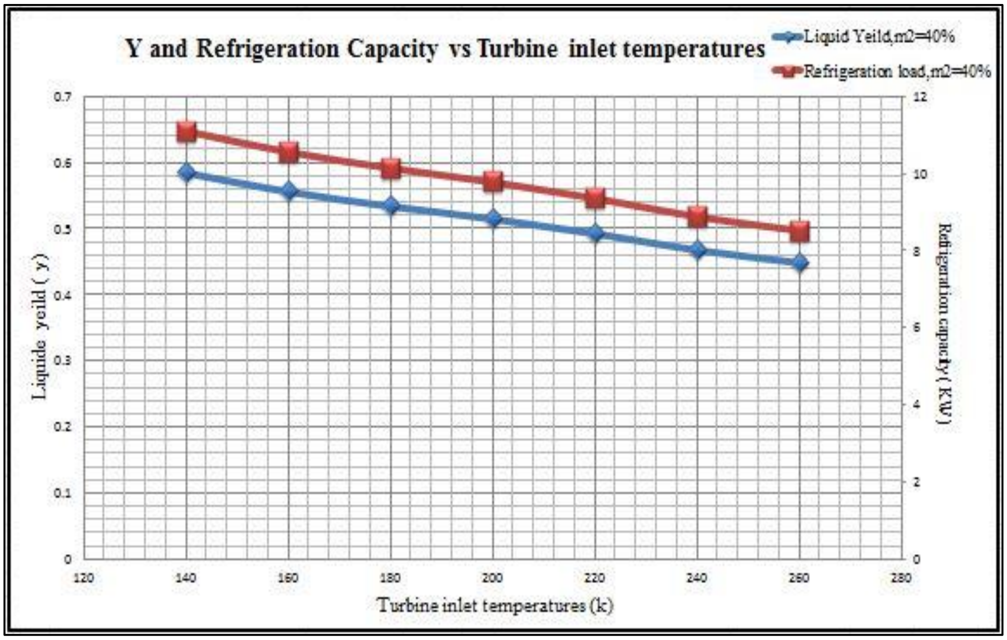


Figure 3.3.1.2: Plot of LN2 production and refrigeration for one Turbine and Three Heat Exchangers with a JT valve: Steady state approach

- Above plots are plotted for the LN2 production and refrigeration load at m2 = 40% at turbine inlet temperatures at 140 K, 160 K, 180 K, 200 K, 220 K, 240 K, 260 K.
- Conclusion from the above plots:
    - ✓ Mass flow rate through a turbine is 40%, at lower turbine inlet temperature (140 K) maximum LN2 production occurs and high refrigeration load is needed and at higher temperatures (260 K) minimum LN2 production and lowest refrigeration load is required.
    - ✓ A point has to be selected where LN2 production increases and refrigeration load decreases.

3.3.2 UA BASED METHOD FOR HEAT EXCHANGER:

This will be based on UA rather than effectiveness otherwise same procedure will be followed as above.

3.3.3 EFFECTIVENESS BASED METHOD ONLY FOR MIDDLE HEAT EXCHANGER:

This will be based on effectiveness of middle heat exchanger otherwise same procedure will be followed as above.

3.4 EFFECT OF COMPRESSOR OUTLET PRESSURE ON LIQUEFACTION AND REFRIGERATION CAPACITY

Compressor outlet pressure is one of the important operating parameter in a given configuration which has to be optimized. A procedure has been developed analytically at given conditions. Developed analytical procedure has been calculated at different compressor outlet pressures e.g. 10 b, 12 b, 14 b,.22b to see its effect on liquefaction at the outlet of JT and refrigeration capacity.

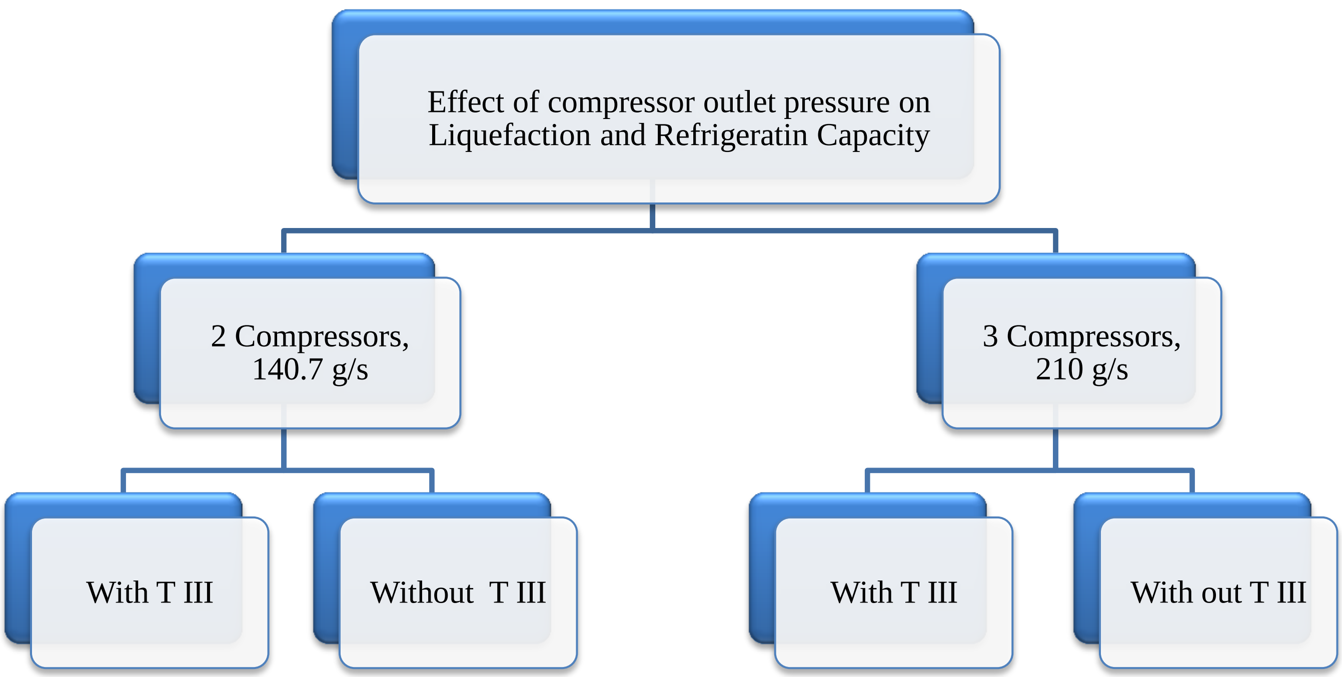


Figure 3.4.1: Effect of compressor outlet pressure on liquid formation at JT outlet and refrigeration capacity

(1) TWO COMPRESSORS AND COMPRESSOR OUTLET MASS FLOW RATE IS 140.7 g/s

(a) With T III :

To achieve cooling capacity of about 2 kW at 4.5 K, total flow rates will be supplied by 3 standard compressors each having delivery helium flow capacity of about 70 g/s. For different operational configurations of having 1, 2 and 3 compressors are analyzed. 2 compressor systems can provide flow rate of 140.7 g/s which is same as that of existing helium plant's nominal compressor flow rate. The cyclic configuration of 2 compressor system is shown below which contains total 8 Heat Exchangers HE 1, HE $LN_{2, \ldots}$ HE VII respectively, Three Turbines in series combination named T I, T II, T III, JT Valve and a Liquefier / Separator where liquefaction rate is 7 g/s. In this 2 compressor system outlet mass flow at the compressor is 140.7 g/s at 14 bar which passes through the hot stream which is denoted as mh and some amount of liquid is formed at the outlet of JT out of which liquefaction rate is removed out for some minor applications and rest is returned back to cold stream as m_c. TS diagram of this configuration has been plotted below and the procedure has been described below with the help of flow chart.

3.4.1 FLOW CHART EXPLANATION FOR TWO COMPRESSOR SYSTEM WITH 3RD TURBINE:

HE II: It is a two stream heat exchanger in which heat has been transferred from hot helium stream to cold helium stream. Thi and Tho is known and minimum approach (θ2) has been assumed such a Tco is calculated from θ2 and Tci from energy balance. By knowing all temperatures UA has been calculated.

HE I: It is a three stream heat exchanger which contains hot and cold stream of helium and another cold stream of GN_2. Minimum approach (θ1) has been assumed to find out helium cold stream outlet temperature which is same as the GN_2 outlet temperature.

HE LN_2: In this HE phase change occur from LN_2 to GN_2 and that heat has been transferred to cool down the helium hot stream temperature to 80 K. Thi is known from HE I calculations. Using all temperatures LN_2 mass flow rate has been found out.

T I: Turbine 1 inlet temperature is user defined which is given as 35.3 K. Turbine expands adiabatically with adiabatic efficiency 76%

HE III: This is a two stream helium heat exchanger in which Thi and Tco is known from HE II. Minimum approach (θ3) has been assumed and Tho, Tci is calculated using energy balance. Knowing all temperatures UA has been found out to know the size of the heat exchanger.

HE IV: It is a three stream helium heat exchanger in which Tci has been calculated from total heat load on hot streams of helium as T II inlet temperature is given as 15.62 K. UA has been calculated from all temperatures.

T II: Turbine 2 expands adiabatically with adiabatic efficiency of 72% with its user defined inlet temperature as 15.62K.

T III: To provide extra cooling effect 3rd turbine is included in this modified Claude cycle which expands adiabatically with adiabatic efficiency of 64% and turbine inlet temperature as a 7.5 K.

HE VII: It is a two stream helium heat exchanger in which Thi is equal to the T III outlet temperature. Minimum approach (θ7) has been assumed to calculate Tco. Tci is given as 4.408 K which is the boiling point of helium. From energy balance Tho has been calculated.

JT Valve: It gives the cooling effect by isenthalpic expansion. Liquid has been formed at the outlet of JT.

HE VI: Minimum approach (θ6) has been assumed between helium hot stream of HE VI and T II outlet stream. It is a two stream helium heat exchanger in which Tco has been calculated from energy balance.

Mixer: Two different inlet streams of helium are mixed together and a product stream's temperature has been calculated using energy balance.

HE V: This is a two stream helium heat exchanger in which all temperatures has been found out and UA is being calculated.

Liquefier / Separator: It is connected after JT valve so that liquid formed at the outlet of JT is accumulated in a liquefier and 7 g/s which are the liquefaction rate is taken out and rest is heated. GHe is sent back to cool the helium hot stream of HE VII.

Refrigeration Capacity of total plant has been calculated.

3.4.2 PLANT LAYOUT FOR GIVEN CONFIGURATION WITH 3RD TURBINE:

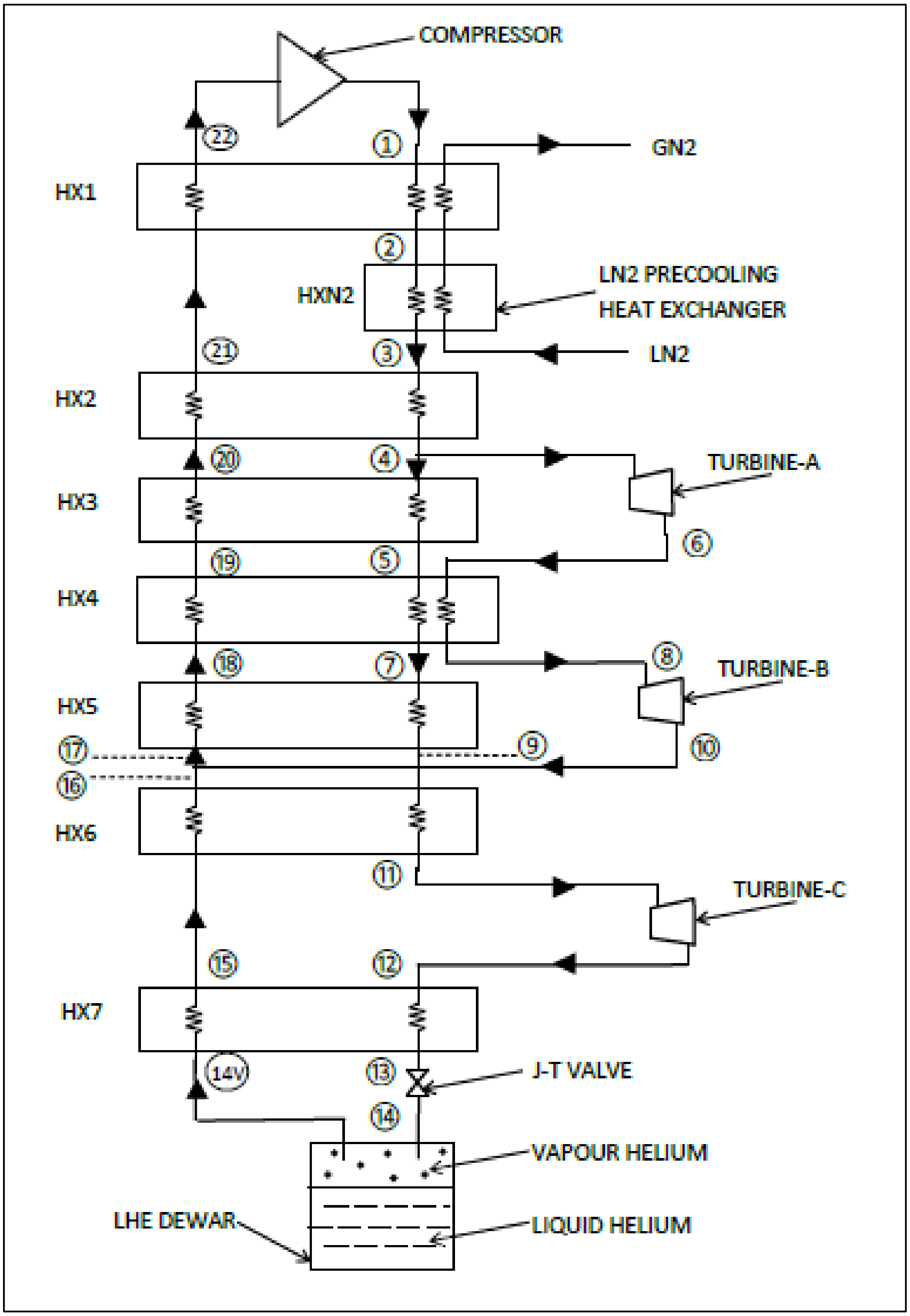


Figure 3.4.2: Plant layout of given configuration with 3rd turbine

3.4.3 TS DIAGRAM OF A GIVEN CONFIGURATION:

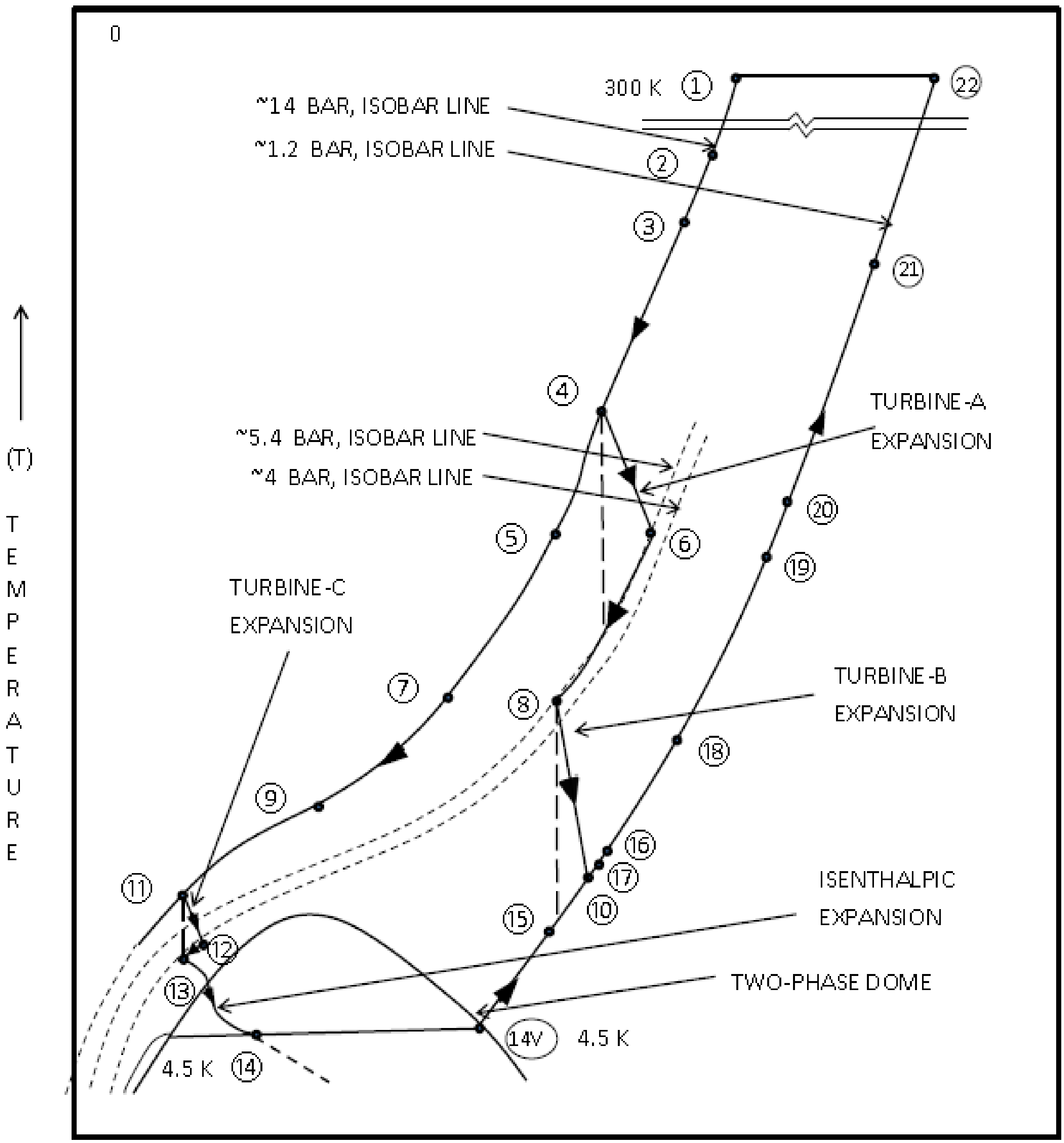
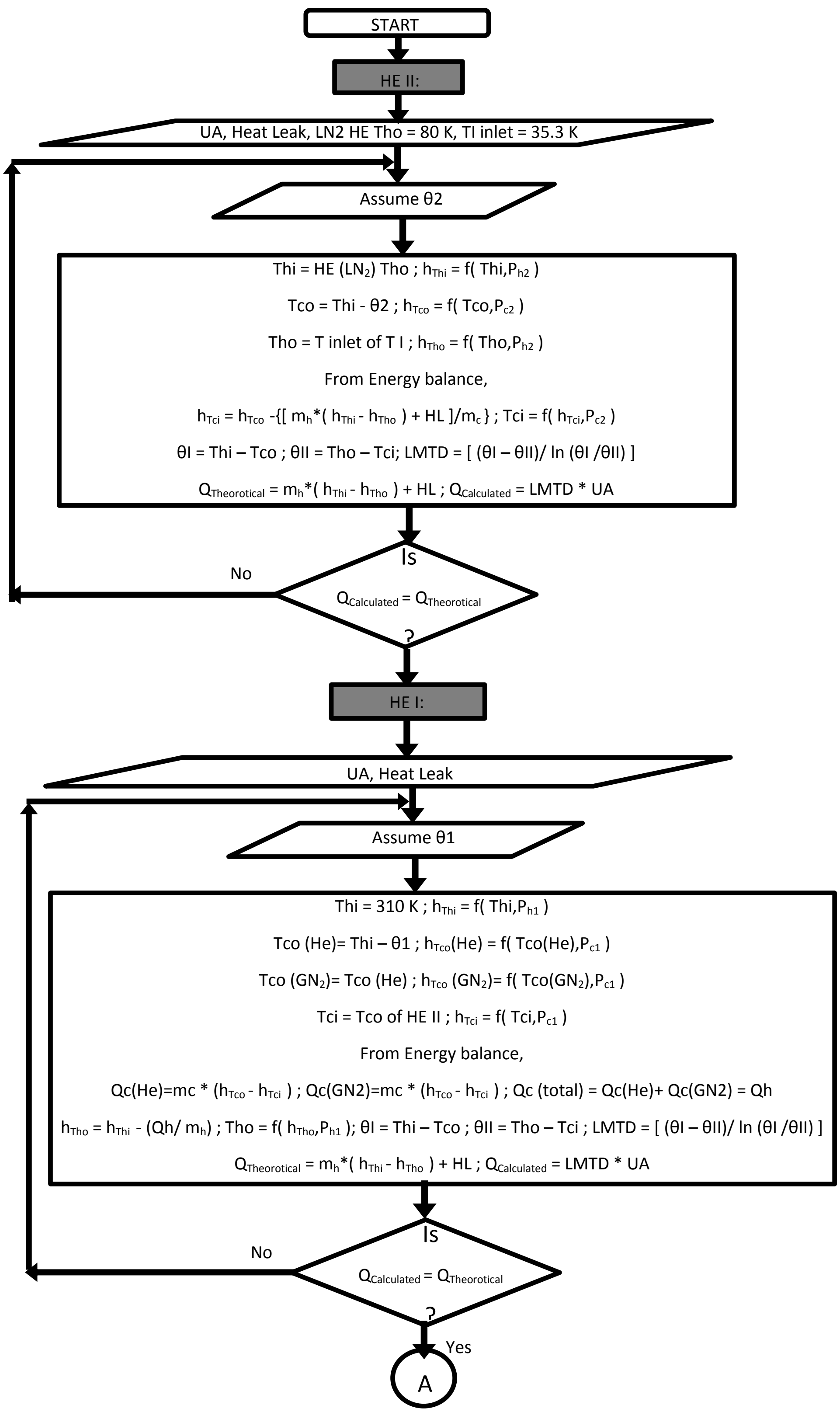
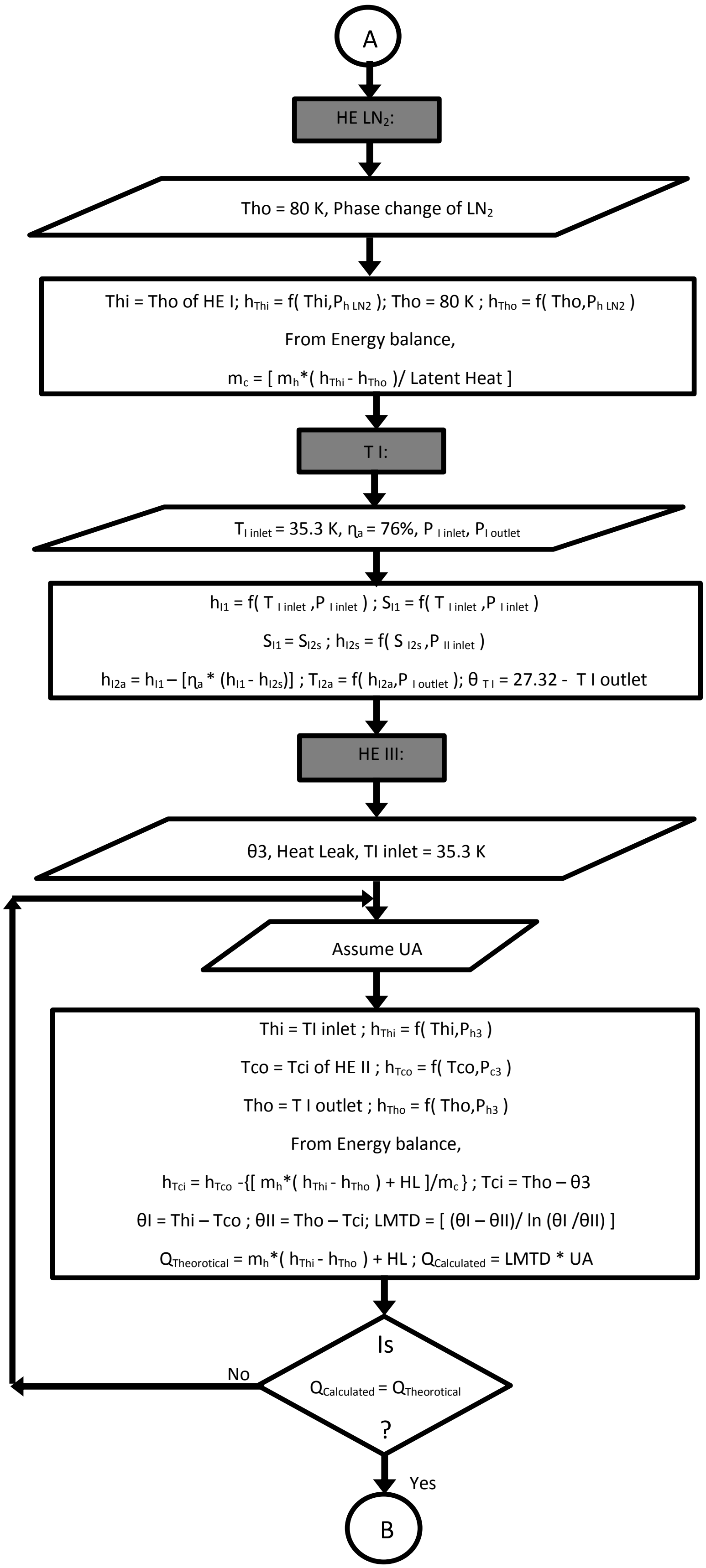
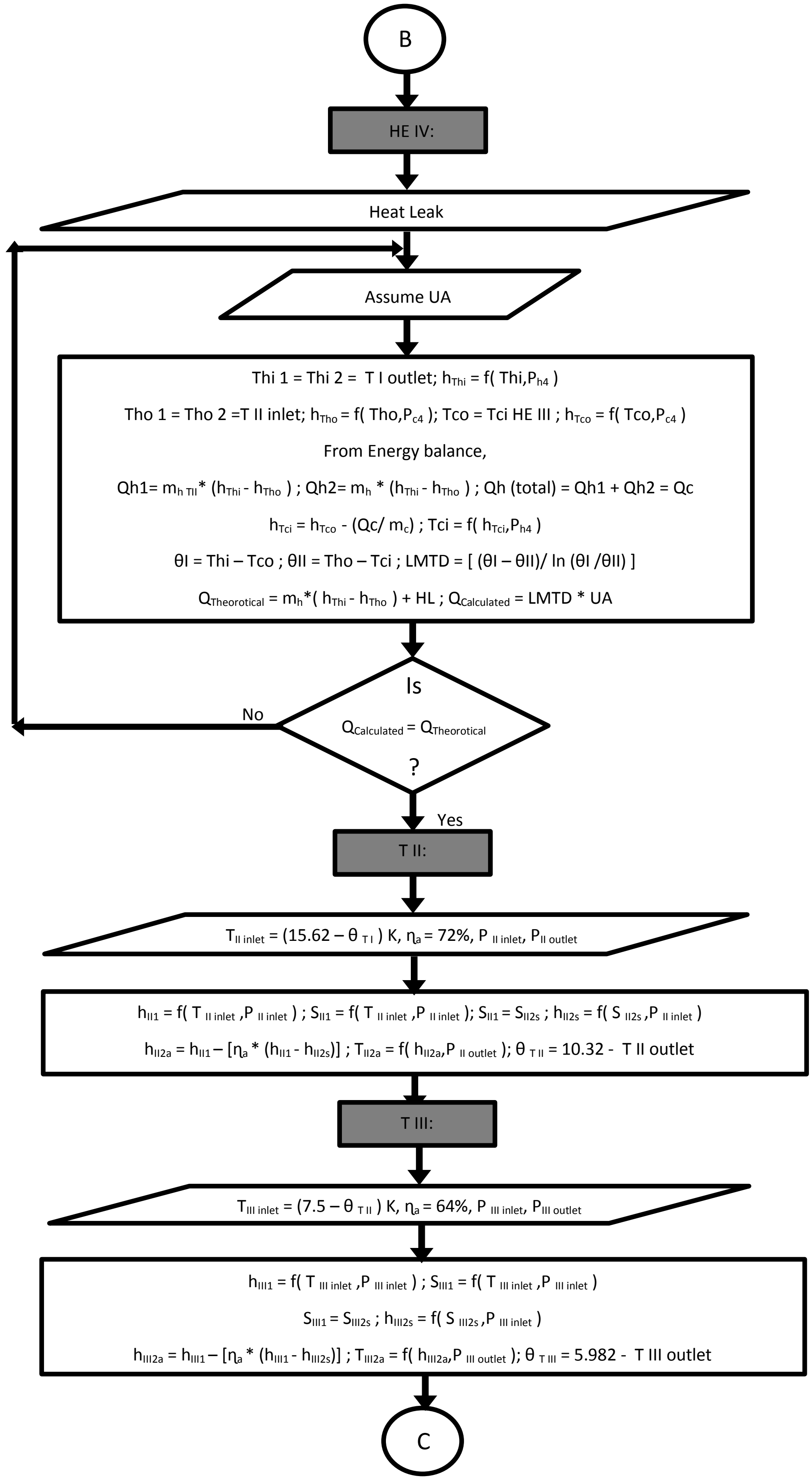


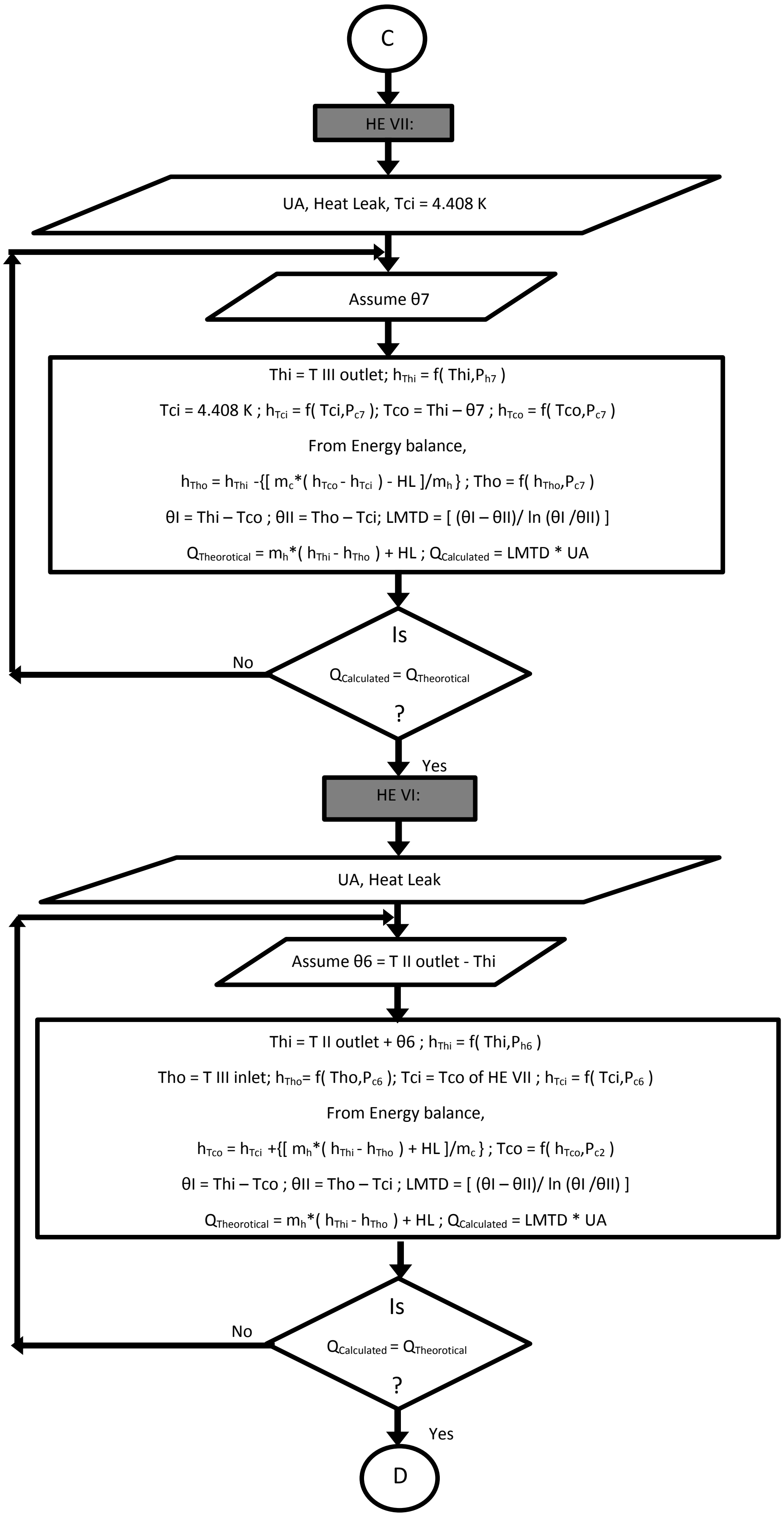
Figure 3.4.3: TS diagram of a given configuration

3.4.4 FLOW CHART FOR 2 COMPRESSORS, 140.7 g/s WITH 3RD TURBINE




A
HE LN_2:
Tho = 80 K, Phase change of LN_2
Thi = Tho of HE I; h_{Thi} = f(Thi,$P_{h\ LN2}$); Tho = 80 K ; h_{Tho} = f(Tho,$P_{h\ LN2}$)
From Energy balance,
m_c = [m_h*(h_{Thi} - h_{Tho})/ Latent Heat]
T I:
$T_{I\ inlet}$ = 35.3 K, η_a = 76%, $P_{I\ inlet}$, $P_{I\ outlet}$
h_{I1} = f($T_{I\ inlet}$,$P_{I\ inlet}$) ; S_{I1} = f($T_{I\ inlet}$,$P_{I\ inlet}$)
S_{I1} = S_{I2s} ; h_{I2s} = f(S_{I2s} ,$P_{II\ inlet}$)
h_{I2a} = h_{I1} – [η_a * (h_{I1} - h_{I2s})] ; T_{I2a} = f(h_{I2a},$P_{I\ outlet}$); $\theta_{T\ I}$ = 27.32 - T I outlet
HE III:
θ3, Heat Leak, TI inlet = 35.3 K
Assume UA
Thi = TI inlet ; h_{Thi} = f(Thi,P_{h3})
Tco = Tci of HE II ; h_{Tco} = f(Tco,P_{c3})
Tho = T I outlet ; h_{Tho} = f(Tho,P_{h3})
From Energy balance,
h_{Tci} = h_{Tco} -{[m_h*(h_{Thi} - h_{Tho}) + HL]/m_c } ; Tci = Tho – θ3
θI = Thi – Tco ; θII = Tho – Tci; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
No
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
Yes
B


B
HE IV:
Heat Leak
Assume UA
Thi 1 = Thi 2 = T I outlet; h_{Thi} = f(Thi,P_{h4})
Tho 1 = Tho 2 =T II inlet; h_{Tho} = f(Tho,P_{c4}); Tco = Tci HE III ; h_{Tco} = f(Tco,P_{c4})
From Energy balance,
Qh1= $m_{h\ TII}$* (h_{Thi} - h_{Tho}) ; Qh2= m_h * (h_{Thi} - h_{Tho}) ; Qh (total) = Qh1 + Qh2 = Qc
h_{Tci} = h_{Tco} - (Qc/ m_c) ; Tci = f(h_{Tci},P_{h4})
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
No
Yes
T II:
$T_{II\ inlet}$ = (15.62 – θ $_{T\ I}$) K, $η_a$ = 72%, P $_{II\ inlet}$, $P_{II\ outlet}$
h_{II1} = f(T $_{II\ inlet}$,P $_{II\ inlet}$) ; S_{II1} = f(T $_{II\ inlet}$,P $_{II\ inlet}$); S_{II1} = S_{II2s} ; h_{II2s} = f(S $_{II2s}$,P $_{II\ inlet}$)
h_{II2a} = h_{II1} – [$η_a$ * (h_{II1} - h_{II2s})] ; T_{II2a} = f(h_{II2a},P $_{II\ outlet}$); θ $_{T\ II}$ = 10.32 - T II outlet
T III:
$T_{III\ inlet}$ = (7.5 – θ $_{T\ II}$) K, $η_a$ = 64%, P $_{III\ inlet}$, $P_{III\ outlet}$
h_{III1} = f(T $_{III\ inlet}$,P $_{III\ inlet}$) ; S_{III1} = f(T $_{III\ inlet}$,P $_{III\ inlet}$)
S_{III1} = S_{III2s} ; h_{III2s} = f(S $_{III2s}$,P $_{III\ inlet}$)
h_{III2a} = h_{III1} – [$η_a$ * (h_{III1} - h_{III2s})] ; T_{III2a} = f(h_{III2a},P $_{III\ outlet}$); θ $_{T\ III}$ = 5.982 - T III outlet
C


C
HE VII:
UA, Heat Leak, Tci = 4.408 K
Assume θ7
Thi = T III outlet; h_{Thi} = f(Thi,P_{h7})
Tci = 4.408 K ; h_{Tci} = f(Tci,P_{c7}); Tco = Thi – θ7 ; h_{Tco} = f(Tco,P_{c7})
From Energy balance,
h_{Tho} = h_{Thi} -{[m_c*(h_{Tco} - h_{Tci}) - HL]/m_h } ; Tho = f(h_{Tho},P_{c7})
θI = Thi – Tco ; θII = Tho – Tci; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
No
Yes
HE VI:
UA, Heat Leak
Assume θ6 = T II outlet - Thi
Thi = T II outlet + θ6 ; h_{Thi} = f(Thi,P_{h6})
Tho = T III inlet; h_{Tho}= f(Tho,P_{c6}); Tci = Tco of HE VII ; h_{Tci} = f(Tci,P_{c6})
From Energy balance,
h_{Tco} = h_{Tci} +{[m_h*(h_{Thi} - h_{Tho}) + HL]/m_c } ; Tco = f(h_{Tco},P_{c2})
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
No
Yes
D

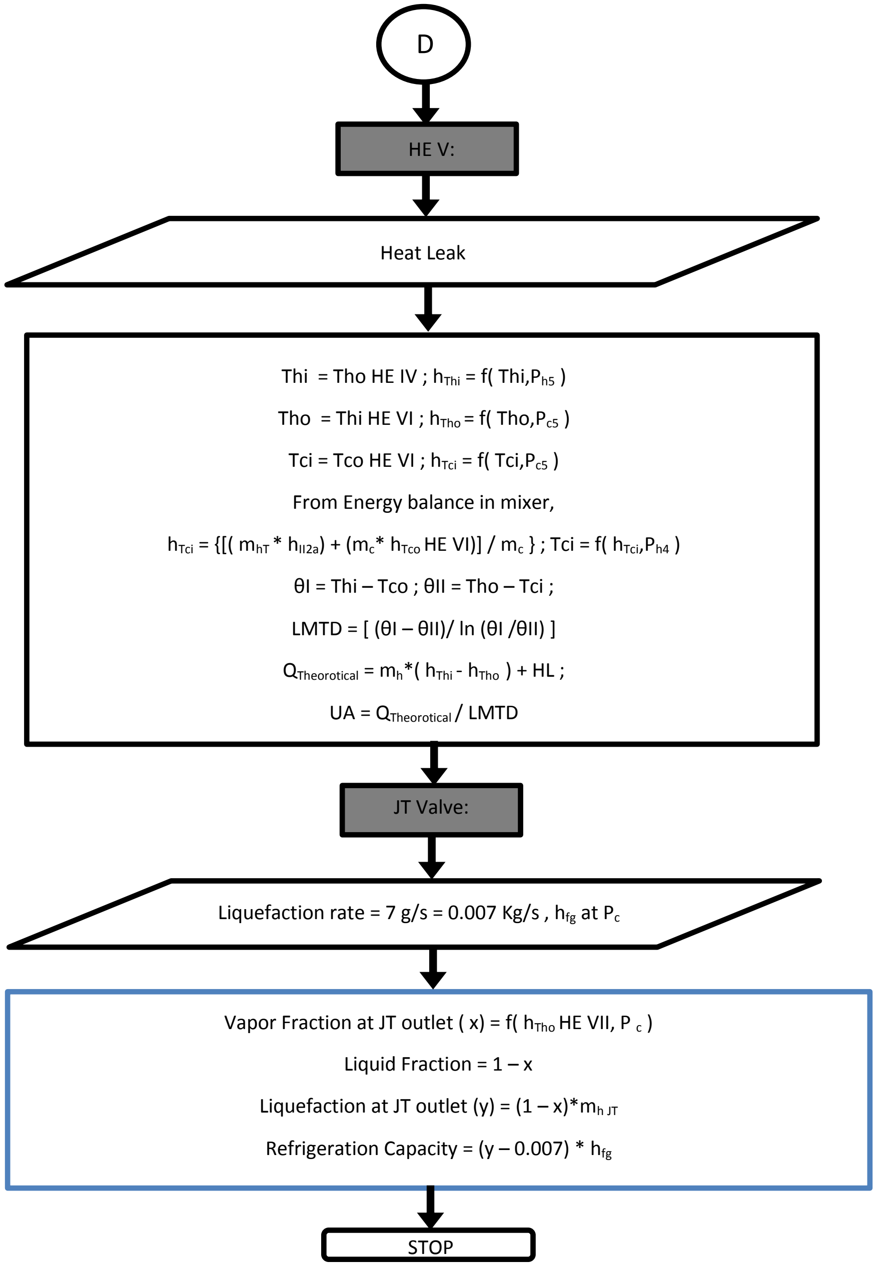


Figure 3.4.4: Flow chart for 2 compressors, 140.7 g/s with 3rd turbine

(b) WITHOUT T III :

This is the same procedure as above but 3rd turbine (T III) has been removed to see the effect of compressor outlet pressure on the liquid formation at the outlet of JT and the refrigeration capacity of plant. The procedure is shown below with the help of flow chart and plant layout diagram.

3.4.5 PLANT LAYOUT OF GIVEN CONFIGURATION WITHOUT 3RD TURBINE:

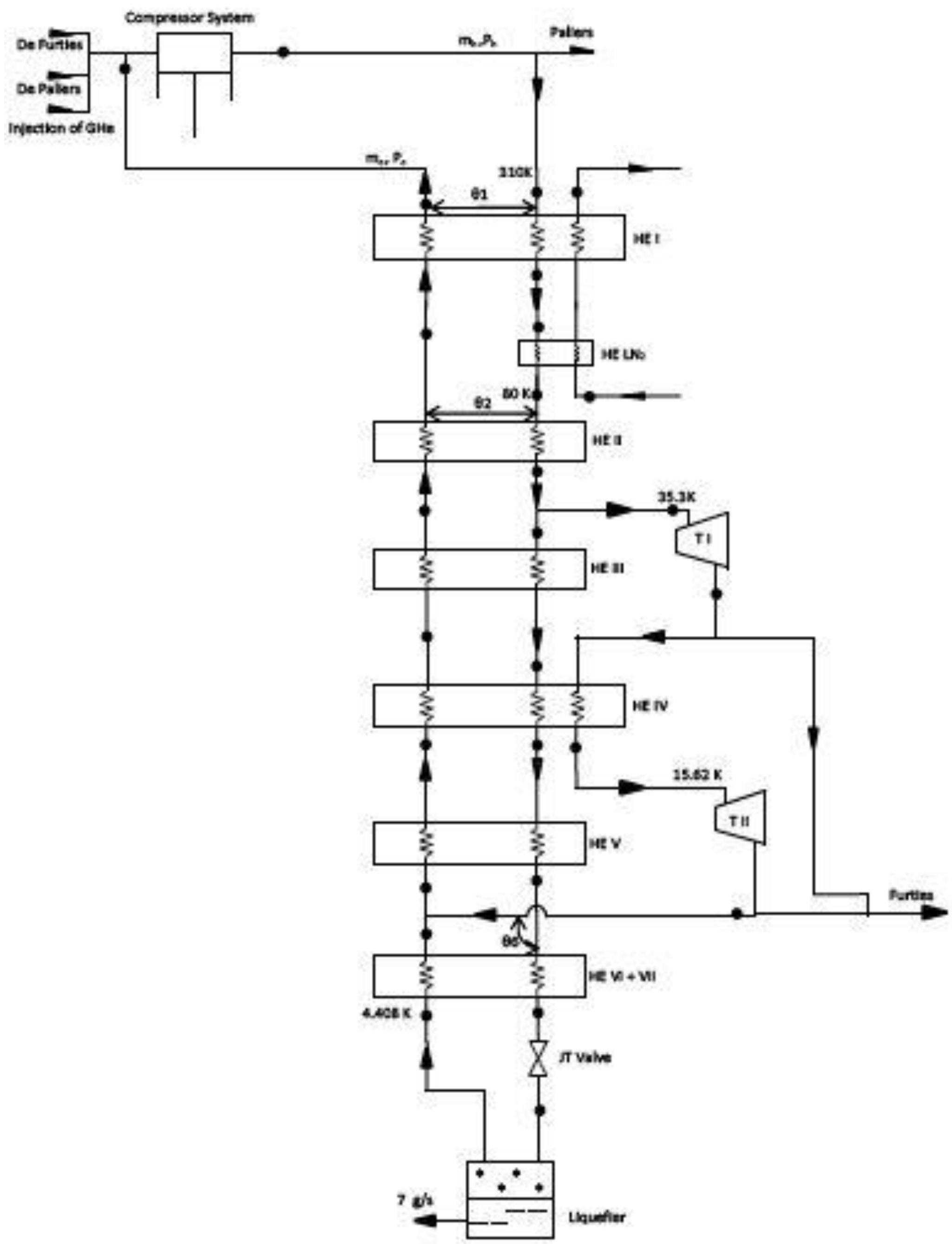
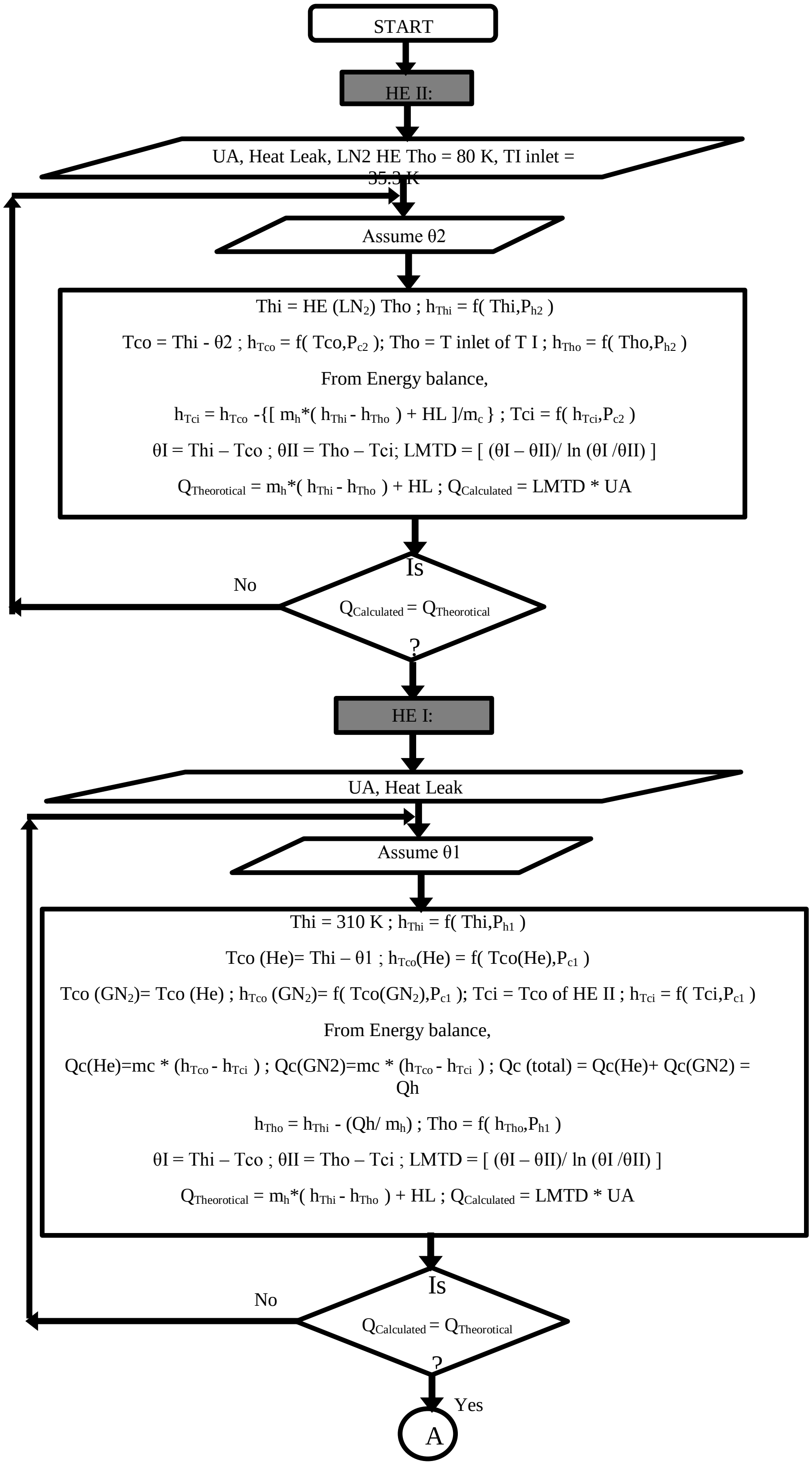
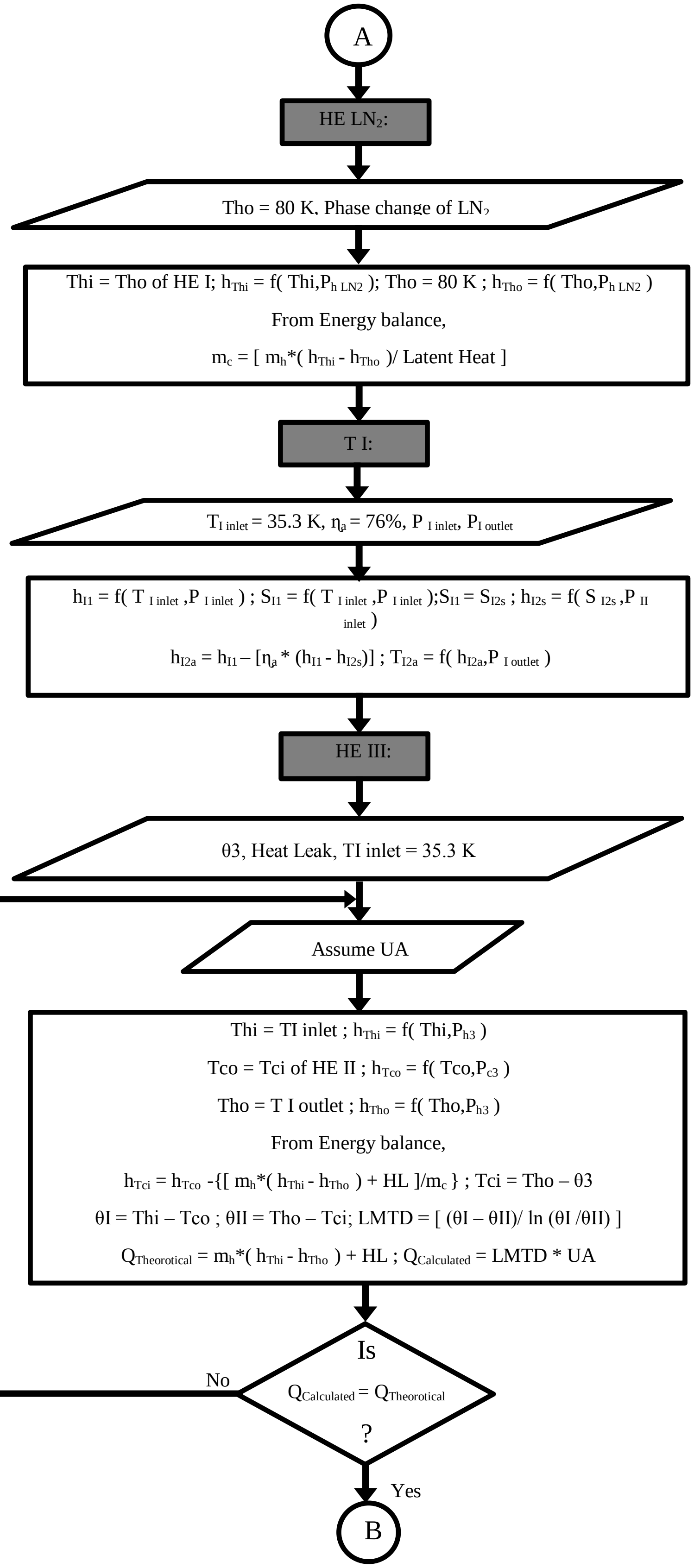
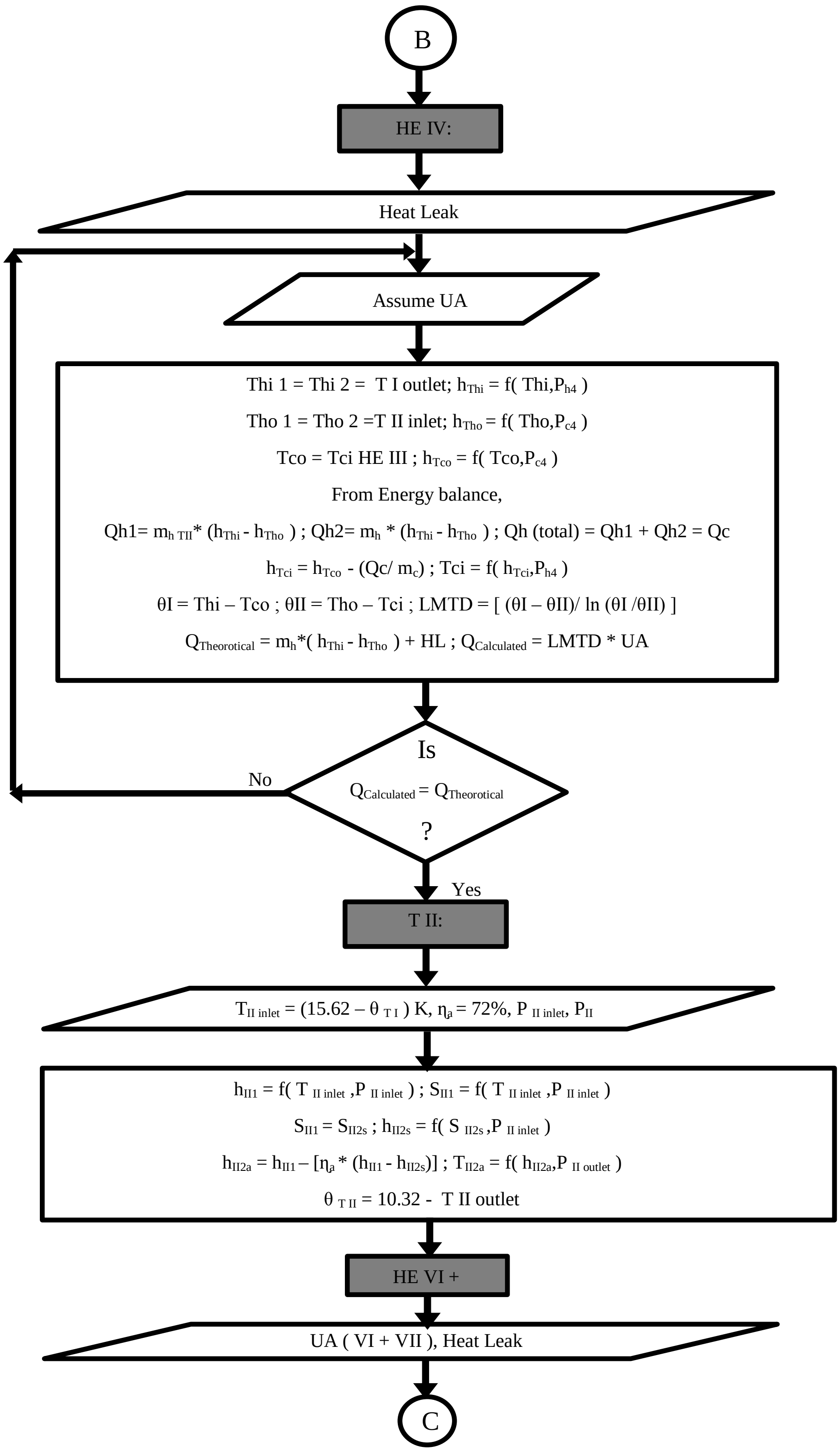


Figure 3.4.5: Plant layout of given configuration without 3rd turbine

3.4.6 FLOW CHART FOR 2 COMPRESSORS, 140.7 g/s WITHOUT 3RD TURBINE




A
HE LN_2:
Tho = 80 K, Phase change of LN_2
Thi = Tho of HE I; h_{Thi} = f(Thi,$P_{h\ LN2}$); Tho = 80 K ; h_{Tho} = f(Tho,$P_{h\ LN2}$)
From Energy balance,
m_c = [m_h*(h_{Thi} - h_{Tho})/ Latent Heat]
T I:
$T_{I\ inlet}$ = 35.3 K, η_a = 76%, $P_{I\ inlet}$, $P_{I\ outlet}$
h_{I1} = f($T_{I\ inlet}$,$P_{I\ inlet}$) ; S_{I1} = f($T_{I\ inlet}$,$P_{I\ inlet}$);S_{I1} = S_{I2s} ; h_{I2s} = f(S_{I2s} ,$P_{II\ inlet}$)
h_{I2a} = h_{I1} – [η_a * (h_{I1} - h_{I2s})] ; T_{I2a} = f(h_{I2a},$P_{I\ outlet}$)
HE III:
θ3, Heat Leak, TI inlet = 35.3 K
Assume UA
Thi = TI inlet ; h_{Thi} = f(Thi,P_{h3})
Tco = Tci of HE II ; h_{Tco} = f(Tco,P_{c3})
Tho = T I outlet ; h_{Tho} = f(Tho,P_{h3})
From Energy balance,
h_{Tci} = h_{Tco} -{[m_h*(h_{Thi} - h_{Tho}) + HL]/m_c } ; Tci = Tho – θ3
θI = Thi – Tco ; θII = Tho – Tci; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
No
Yes
B


B
HE IV:
Heat Leak
Assume UA
Thi 1 = Thi 2 = T I outlet; h_{Thi} = f(Thi,P_{h4})
Tho 1 = Tho 2 =T II inlet; h_{Tho} = f(Tho,P_{c4})
Tco = Tci HE III ; h_{Tco} = f(Tco,P_{c4})
From Energy balance,
Qh1= $m_{h\ TII}$* (h_{Thi} - h_{Tho}) ; Qh2= m_h * (h_{Thi} - h_{Tho}) ; Qh (total) = Qh1 + Qh2 = Qc
h_{Tci} = h_{Tco} - (Qc/ m_c) ; Tci = f(h_{Tci},P_{h4})
θI = Thi – Tco ; θII = Tho – Tci ; LMTD = [(θI – θII)/ ln (θI /θII)]
$Q_{Theorotical}$ = m_h*(h_{Thi} - h_{Tho}) + HL ; $Q_{Calculated}$ = LMTD * UA
No
Is
$Q_{Calculated}$ = $Q_{Theorotical}$
?
Yes
T II:
$T_{II\ inlet}$ = (15.62 – $\theta_{T\ I}$) K, η_a = 72%, $P_{II\ inlet}$, P_{II}
h_{II1} = f($T_{II\ inlet}$,$P_{II\ inlet}$) ; S_{II1} = f($T_{II\ inlet}$,$P_{II\ inlet}$)
S_{II1} = S_{II2s} ; h_{II2s} = f(S_{II2s} ,$P_{II\ inlet}$)
h_{II2a} = h_{II1} – [η_a * (h_{II1} - h_{II2s})] ; T_{II2a} = f(h_{II2a},$P_{II\ outlet}$)
$\theta_{T\ II}$ = 10.32 - T II outlet
HE VI +
UA (VI + VII), Heat Leak
C

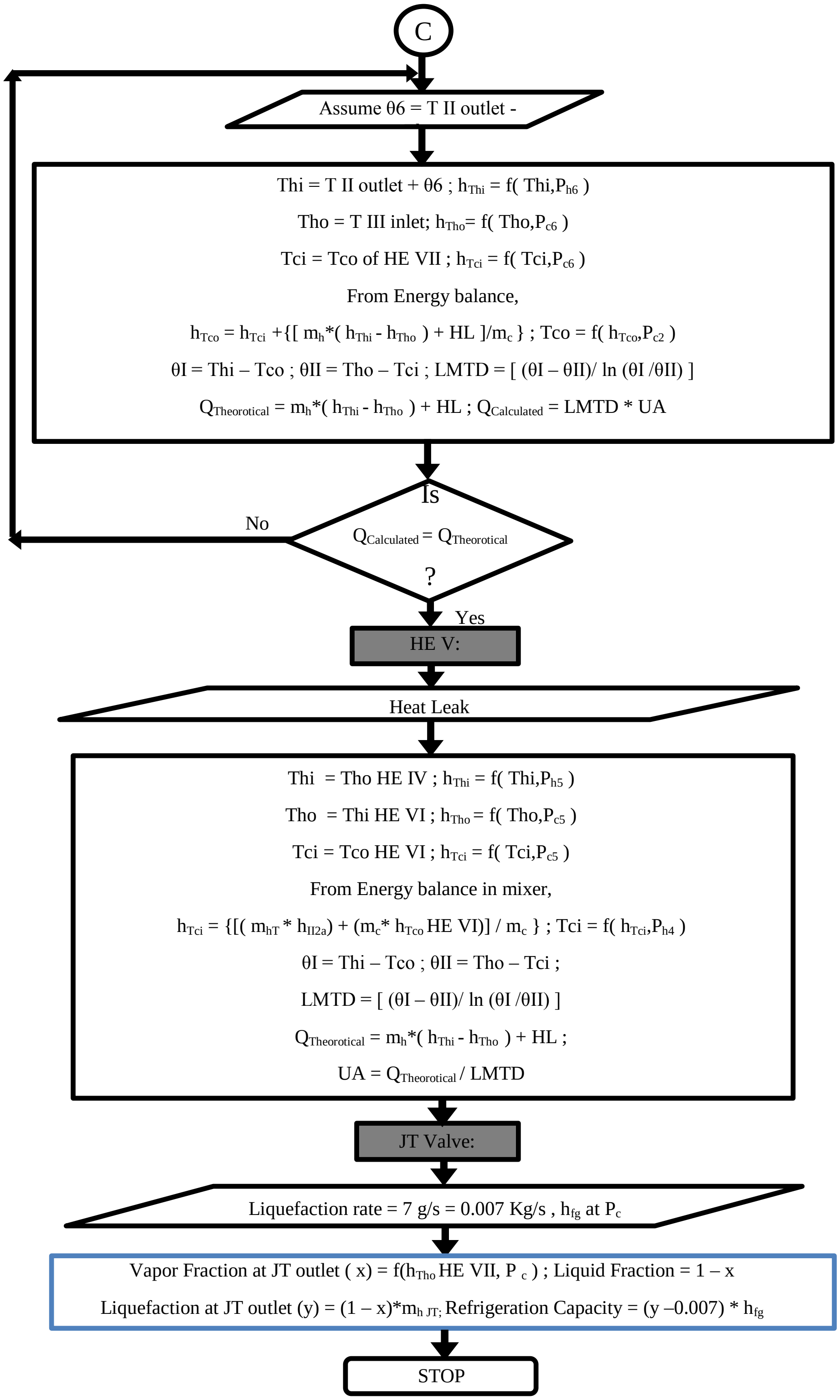


Figure 3.4.6: Flow chart for 2 compressors, 140.7 g/s without 3rd turbine

(2) THREE COMPRESSORS AND COMPRESSOR OUTLET MASS FLOW RATE IS 210 g/s

(a) With T III :

Same procedure has been adopted as above with 3rd turbine using compressor outlet mass flow rate of 210 g/s instead of 140.7 g/s and the whole cycle calculations has been done analytically at different compressor outlet pressures.

(b) Without T III :

The only change in compressor outlet mass flow rate has been done from 140.7 g/s to 210 g/s as compressor system contains 3 compressors at different compressor outlet pressures and the results has been plotted to see how refrigeration capacity of plant varies.

Chapter-4

RESULTS AND DISCUSSION

4.1 EFFECT OF COMPRESSOR OUTLET PRESSURE ON A GIVEN CONFIGURATION:

(1) TWO COMPRESSOR SYSTEM WITH OUTLET MASS FLOW RATE IS 140.7 g/s:

(a) With T III:

Analytically developed procedure for 2 compressors system with outlet mass flow rate of 140.7 g/s is at 14 bar compressor outlet pressure. At different compressor outlet pressures liquid formation at JT outlet and refrigeration capacity is calculated and tabulated below. Graph of liquid formation at JT outlet and refrigeration capacity against compressor outlet pressure has been plotted below which shows that both liquid formation and refrigeration capacity increases as compressor outlet pressure is increasing.

Pressure	JT Inlet	liquefaction	Refrigeration load
Pa	k	g/s	W
1200000	5.761282527	29.33742534	436.55
1400000	5.255432267	42.91480427	701.90
1600000	4.885541404	49.2079059	824.89
1800000	4.687424143	52.03264347	880.10
2000000	4.583452649	53.40708397	906.96
2200000	4.525941217	54.13970546	921.28

Table 4.1.1: Liquid formation at JT outlet, Refrigeration capacity and JT inlet temperature at different compressor outlet pressure for 2 compressor system with 3rd turbine

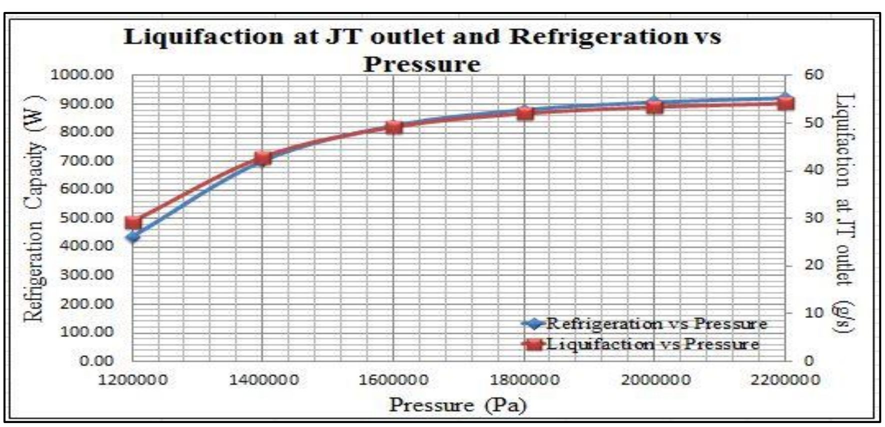


Figure 4.1.1: Liquid formation at JT outlet, Refrigeration capacity VS pressure for 2 compressor system with 3rd turbine

(b) Without T III:

As per discussion in 3.4.5, at different compressor outlet pressures liquid formation at JT outlet and refrigeration capacity is calculated and tabulated below. Graphs has been plotted against compressor outlet pressure. Both the plots shows increasing nature till pressure reaches to 17.1 bar and then decreases. Maximum value of liquid formation at JT outlet is 36.7 g/s and Refrigeration capacity is 582.12 W. at 17.1 bar compressor outlet pressure.

Pressure Pa	JT Inlet k	liquefaction g/s	Refrigeration load W
1200000	6.74630532	16.98811153	195.20
1400000	5.787061827	28.97207206	429.41
1600000	4.894978169	35.90612238	564.93
1650000	4.749956438	36.53881825	577.29
1680000	4.68102954	36.72474613	580.93
1700000	4.641969391	36.77876641	581.98
1705000	4.633107624	36.78343028	582.07
1710000	4.624486816	36.78567726	582.12
1715000	4.616176123	36.7849161	582.10
1720000	4.608161368	36.78129384	582.03
1750000	4.565839724	36.70448253	580.53
1800000	4.514004042	36.4018711	574.62
1900000	4.456524464	35.37925816	554.63
2000000	4.431219959	34.07452	529.13
2200000	4.414242356	31.17803007	472.53

Table 4.1.2: Liquid formation at JT outlet, Refrigeration capacity and JT inlet temperature at different compressor outlet pressure for 2 compressor system without 3rd turbine

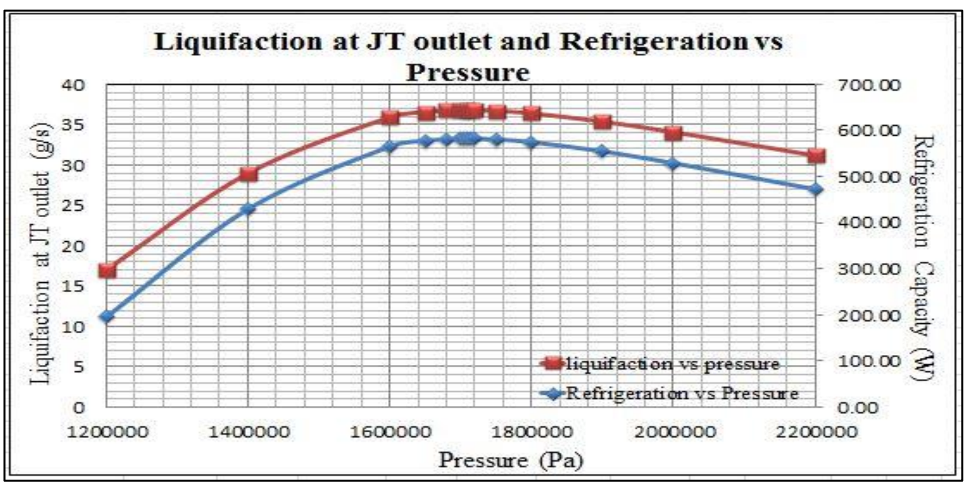


Figure 4.1.2: Liquid formation at JT outlet, Refrigeration capacity VS pressure for 2 compressor system without 3rd turbine

(2) THREE COMPRESSOR SYSTEM WITH OUTLET MASS FLOW RATE IS 210 g/s:

(a) With T III:

Below plot shows that liquid formation at JT outlet and refrigeration capacity is directly proportional to pressure. It increases as compressor outlet pressure increases.

pressure	JT Inlet	liquefaction	Refrigeration load
Pa	k	g/s	W
1100000	5.927323236	32.67443164	501.77
1200000	5.708255454	46.71577453	776.19
1300000	5.441330461	58.11690347	999.01
1400000	5.182552054	66.10965052	1155.21
1500000	4.974039679	71.40626042	1258.73
1600000	4.821330897	74.85983351	1326.22
1700000	4.713808133	77.12370443	1370.47
1800000	4.638507297	78.63827585	1400.07
1900000	4.585195802	79.67848098	1420.40
2000000	4.546826422	80.41167099	1434.73
2100000	4.51873756	80.94054319	1445.06
2200000	4.497854014	81.32956259	1452.67

Table 4.1.3: Liquid formation at JT outlet, Refrigeration capacity and JT inlet temperature at different compressor outlet pressure for 3 compressor system with 3rd turbine

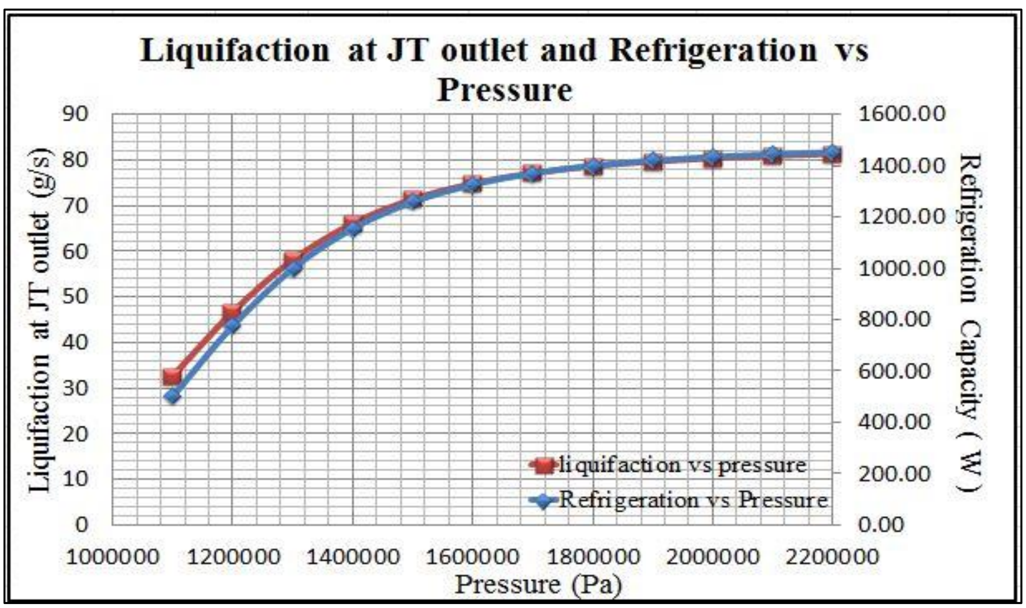


Figure 4.1.3: Liquid formation at JT outlet, Refrigeration capacity VS pressure for 3 compressor system with 3rd turbine

(b) Without T III:

Refrigeration capacity and liquid production shows a peak value at 16 bar and then decreases.Maximum values of refrigeration capacity is 983.64 W and liquid formation at JT outlet is 57.33 g/s at 16 bar compressor outlet pressure.

Pressure Pa	JT Inlet k	liquefaction g/s	Refrigeration load W
1200000	6.380853736	34.06260995	528.90
1400000	5.381981306	50.21483081	844.57
1500000	4.927399245	55.24397429	942.86
1600000	4.624491574	57.3304184	983.64
1700000	4.482354176	56.9745864	976.68
1800000	4.432108822	55.36258769	945.18
1900000	4.416155075	53.3020345	904.91
2000000	4.410984197	51.10249639	861.92

Table 4.1.4: Liquid formation at JT outlet, Refrigeration capacity and JT inlet temperature at different compressor outlet pressure for 3 compressor system without 3rd turbine

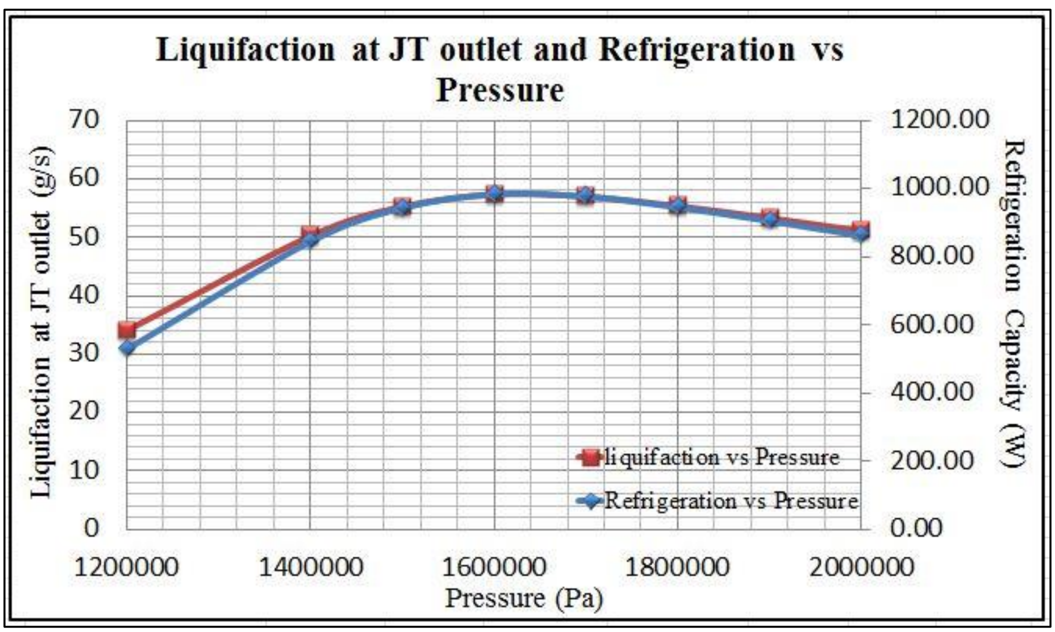


Figure 4.1.4: Liquid formation at JT outlet, Refrigeration capacity VS pressure for 3 compressor system without 3rd turbine

4.2 EFFECT OF COMPRESSOR OUTLET MASS FLOW RATE ON A GIVEN CONFIGURATION:

(1) TWO COMPRESSOR SYSTEM WITHOUT 3RD TURBINE:

A procedure has been developed such that process fluid temperature for component entry and exit points have been kept constant for different compressor flow rates (however they will change a bit) and change the UA of each Heat Exchanger with different compressor outlet mass flow rates. This procedure has been developed for 2 compressor system without 3rd turbine. Below table shows the variation of liquid formation at JT outlet and refrigeration capacity with the different compressor outlet mass flow rate. Plot shows as a compressor outlet mass flow increases which give the highest liquid formation at JT outlet and highest refrigeration capacity at that point. Bigger is the system, higher efficiency with higher mass flow rate at compressor outlet.

Compressor outlet mass flow rate	liquid formation at JT outlet	Refrigeration Capacity
g/s	g/s	W
210	68.52496711	1202.42
180	43.38479234	711.09
140.7	28.78552213	425.77
100	15.87540648	173.46
70	8.262593718	24.68

Table 4.2.1: Liquid formation at JT outlet, Refrigeration capacity at different compressor outlet mass flow rate for 2 compressor system without 3rd turbine

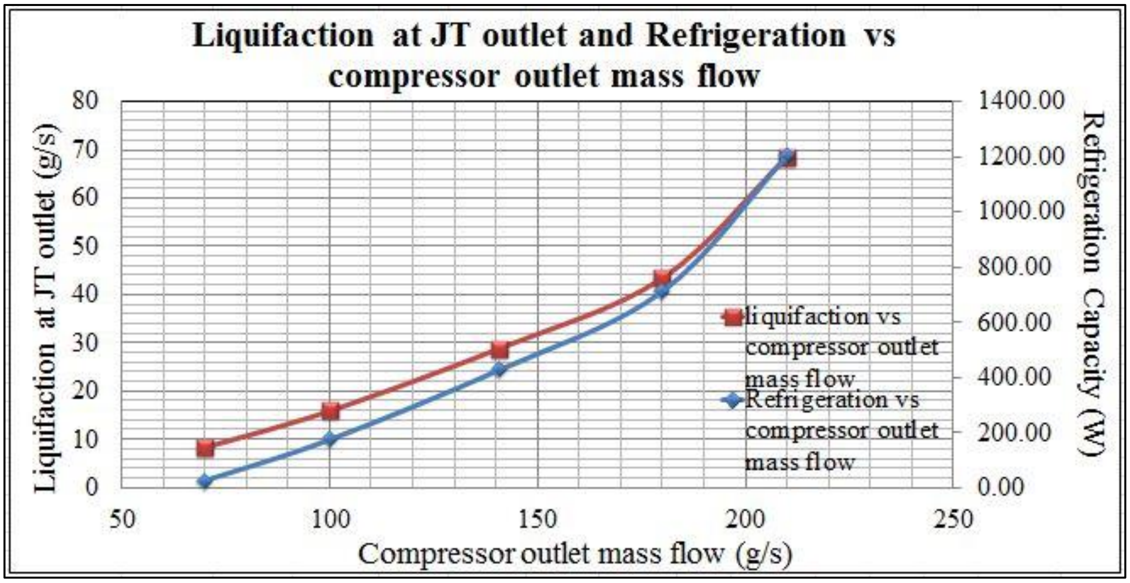


Figure 4.2.1: Liquid formation at JT outlet, Refrigeration capacity at different compressor outlet mass flow rate for 2 compressor system without 3rd turbine

Chapter-5

VALIDATION USING ASPEN HYSYS

5.1 INTRODUCTION TO ASPEN HYSYS:

It belongs to aspen one engineering family. Aspen one is a model processing simulation software by Aspen Tech. Its main purpose is to simulate the optimized configuration with operational excellence.

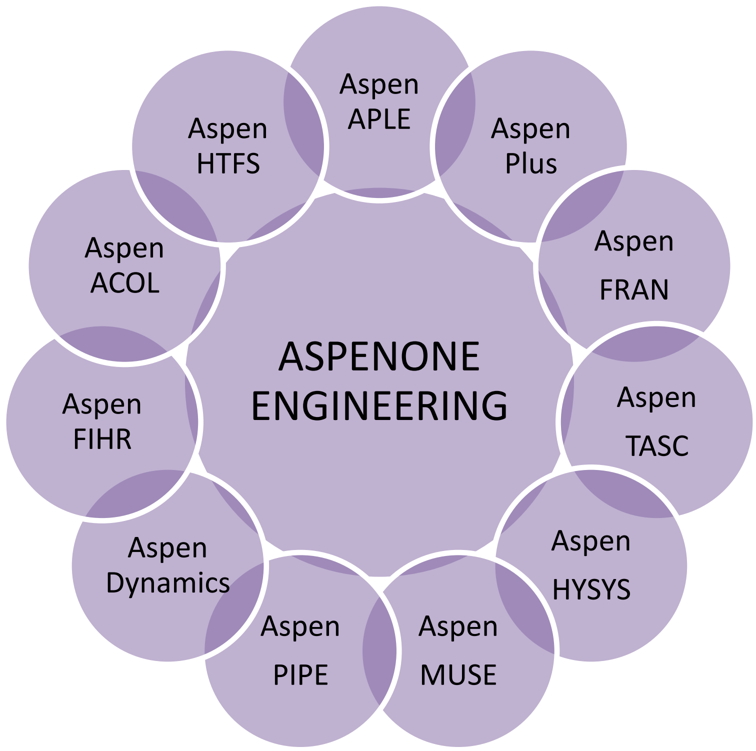


Figure 5.1.1: Aspen ONE Engineering Family

Aspen HYSYS consists of three process modeling regions: Gas processing, Refining, Chemicals

1. Gas Processing

- **Steady State:** It models a gas refrigeration plant consisting of compressor, expander, gas heat exchanger, chiller, low-temperature separator etc.
- **Dynamics:** All models are built up in dynamic processing mode.

2. Refining

- **Steady State:** This modeling includes a crude oil processing facility consisting of a pre-flash drum, crude furnace and an atmospheric crude column.
- **Dynamics:** Models the Refining problem in Dynamic mode.

3. Chemicals

- **Steady State:** It contains modeling of a propylene glycol production process consisting of a continuously-stirred-tank reactor and a distillation tower.
- **Dynamics:** Models the Chemical problem in Dynamic mode.

Helium liquefaction system comes under gas process modeling in a steady state.

5.2 ENTERING THE SIMULATION ENVIRONMENT:

For entering into the Simulation environment, detailed instructions for choosing a property package and components, installing and defining streams, unit operations, and using various aspects of the HYSYS interface to examine the results while you are creating the simulation. The gas processing simulation is built using following basic steps:

1. Create a unit set.
2. Choose a property package.
3. Select the components.
4. Create and specify the feed streams.
5. Install and define the unit operations prior to the Heat Exchanger / Compressor / Turbine.
6. Install and define the Heat Exchanger / Compressor / Turbine.

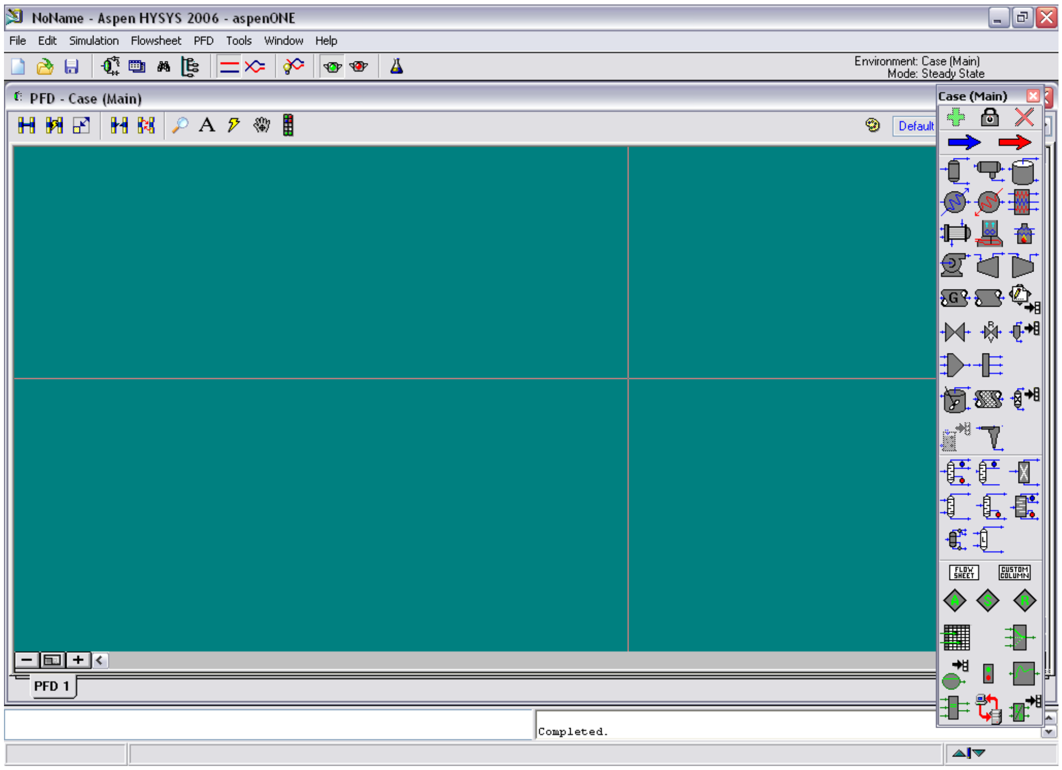


Figure 5.2.1: Aspen HYSYS Simulation Environment

5.3 PROCESS DESIGN PROCEDURE IN ASPEN HYSYS:

For creating a new case in Aspen HYSYS, select New from file menu and Case from submenu. Then Simulation Basis Manager Window will appear. The Simulation Basis Manager is the main property view of the Simulation environment. You can access information in Simulation Basis manager while the other areas of HYSYS are kept on hold avoiding unnecessary Flow sheet calculations. Any changes made in the Simulation Basis environment will take effect at the same time. At the same time thermodynamic data has been fixed and cannot be manipulated in the flow sheet of simulation environment.

The minimum input criteria of Simulation Basis manager are to select a Fluid Package with an attached Property Package and At least one component in the Fluid Package. In a simulation basis manager all components are present in a component manager tab which contains all chemical information of respective component. This information is stored as component list from the collection of library. The Components Manager always contains a Master Component List that cannot be deleted.

This master list contains every component available from all component lists. If you add components to any other component list, they automatically get added to the Master Component List. Also, if you delete a component from the master, it also gets deleted from any other component list that is using that component. Fluid Package contains all necessary information of a component which is required in Calculation. There are four key advantages to this approach:

- All associated information is defined in a single location, allowing for easy creation and modification of the information.
- Fluid Packages can be exported and imported as completely defined packages for use in any simulation.
- Fluid Packages can be cloned, which simplifies the task of making small changes to a complex Fluid Package.
- Multiple Fluid Packages can be used in the same simulation.

In simulation Basis Manager contains fluid package tab on fluid package manager. In this tab many other fluid packages can be created as well as manipulated. Once we go to fluid package manager then we can choose one appropriate fluid package according to our requirement of properties of given configuration. Selected fluid package from fluid package manager in Simulation Basis Environment is listed in the Current Fluid packages group with the following information: name, number of components attached to the fluid package, and property package attached to the fluid package. To see the list of all fluid packages click view from the Fluid Package tab of the Simulation Basis Manager and Add button to add respective Fluid Package to the Environment. Select the proper fluid package and from the Component List Selection drop-down list, select the required components for simulation of given configuration.

From Fluid package manager, active the Aspen properties tab and select the required Fluid Package. Here RefProp fluid package has been selected for a given configuration and from Aspen properties database two components i.e. Helium-4, Nitrogen is selected.

After selecting fluid packages and components, click on Enter Simulation Environment button so that a process flow sheet window will appear. From Menu bar, set the preferences in Tools option.

Choose a Unit set from variables in the preferences or add a new user defines unit set. In process flow sheet unit operations can be installed in many ways. There are many unit operations available on a palette. As soon as we double click on the unit operations to be installed tab opens. There we can input all the connections and values. In worksheet of that unit operation we can see the material streams information which is automatically calculated as soon as we enter some input values. We can reposition streams and operations. In steady state analysis recycler unit operations can be used to calculate the unknown parameters in the process flow diagram. The process flow diagram (PFD) provides the best representation of the flow sheet. Using the PFD gives you immediate reference to the progress of the simulation currently being built, such as what streams and operations are installed, flow sheet connectivity, and the status of objects. In addition to graphical representation, we can build our flow sheet within the PFD.

5.4 INPUT PARAMETERS IN A PFD:

From simulation basis manager, aspen properties database two components Nitrogen and Helium-4 are taken as material stream. From fluid package, Aspen properties, Riprap is selected as a fluid package. Then enter into the simulation environment. All unit operations are arranged in order and linked by material streams. PFD is shown below for a given configuration in HYSYS:

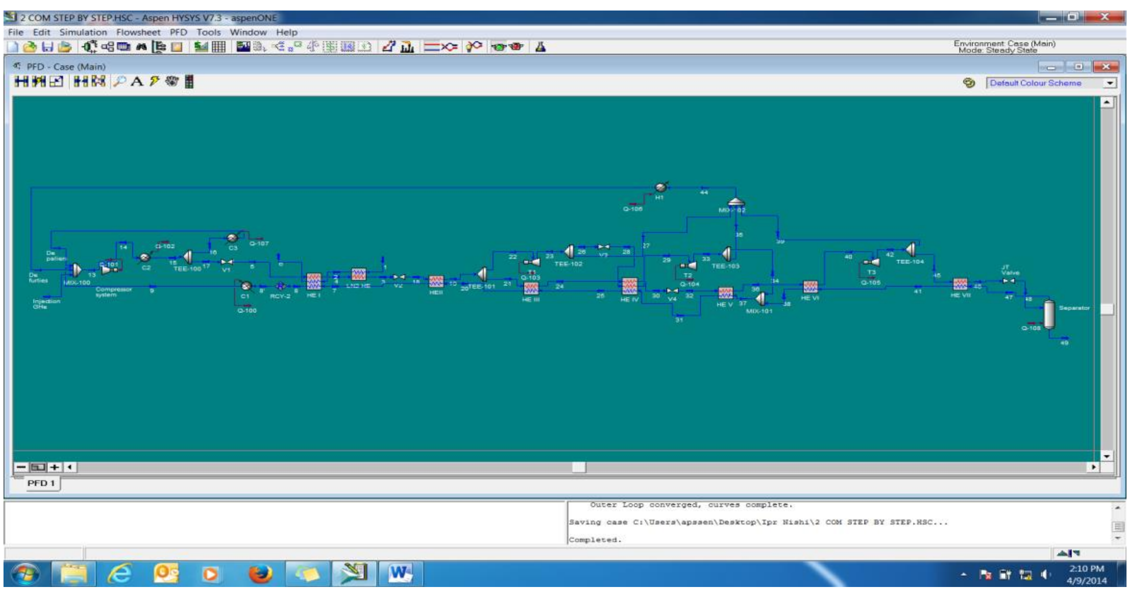


Figure 5.4.1: PFD for a given configuration in a simulation environment

For each unit operations following input values are given.

1. Cooler (C1)

   Outlet temperature = 300 K

   Pressure drop = 0.15 bar

2. Mixer (MIX 100)

   Inlet streams:

   - De Furties:

     Inlet temperature = 300 K

     Inlet pressure = 1.05 bar

     Mass flow rate = 0.7 g/s

   - De Paliers:

     Inlet temperature = 300 K

     Inlet pressure = 1.05 bar

     Mass flow rate = 8 g/s

   - Injection of GHe:

     Inlet temperature = 300 K

     Inlet pressure = 1.05 bar

     Mass flow rate = 7 g/s

3. Compressor system

   Mass flow rate = 140.7 g/s

   Inlet temperature = 300 K

   Inlet pressure = 1.05 bar

   Outlet pressure =14 bar

4. Cooler (C2)

   Outlet temperature = 310 K

   Pressure drop = 0.2 bar

5. Tee (Tee 100)

   Mass flow rate to Paliers = 8 g/s

6. Cooler (C3)

   Outlet temperature = 300 K

   Pressure drop = 12.75 bar

7. Valve (V1)

   Pressure drop = 0.5 bar

8. Heat exchanger 1 (HE I)

   Pressure drop in hot stream = 0.1 bar

   Pressure drop in Cold stream = 0 bar

   Temperature at 8 = 307 K

   Minimum Approach ($\theta 1$) = 3 K

   Heat Leak = 0.07 KW

9. Heat exchanger LN_2 (HE LN_2)

   Pressure drop in hot stream = 0 bar

   Pressure drop in Cold stream = 0 bar

   LN_2 inlet vapor fraction = 0.00

   LN_2 outlet vapor fraction = 1.00

   Inlet temperature = 79.19 K

   LN_2 mass flow rate = 23.96 g/s

10. Valve (V2)

    Pressure drop = 0.15 bar

11. Heat exchanger 2 (HE II)

    Pressure drop in hot stream = 0.06 bar

    Pressure drop in Cold stream = 0 bar

    Minimum Approach ($\theta 2$) = 1.344 K

    Heat Leak = 0.05 KW

12. Tee (Tee 101)

    Mass flow rate to T I = 74.92 g/s

13. Turbine 1 (T I)

    Efficiency of turbo expander = 76 %

    Outlet pressure = 5.4 bar

14. Heat exchanger 3 (HE III)

    Pressure drop in hot stream = 0 bar

    Pressure drop in Cold stream = 0 bar

    Minimum Approach ($\theta 3$) = 0.527 K

Heat Leak = 0.015 KW

UA = 1372 W/C

15. Tee (Tee 102)

    Mass flow rate to De Furties = 0.2 g/s

16. Valve (V3)

    Pressure drop = 0.1 bar

17. Heat exchanger 4 (HE IV)

    Pressure drop in hot stream = 0 bar

    Pressure drop in Cold stream = 0 bar

    Heat Leak = 0.04 KW

    Temperature at 29 and 30 = 15.62 K

18. Valve (V4)

    Pressure drop = 0.15 bar

19. Turbine 2 (T II)

    Efficiency of turbo expander = 72 %

    Outlet pressure = 1.2 bar

20. Tee (Tee 103)

    Mass flow rate to De Furties = 0.2 g/s

21. Heat exchanger 5 (HE V)

    Pressure drop in hot stream = 0 bar

    Pressure drop in Cold stream = 0 bar

    Heat Leak = 0.025 KW

    Temperature at 36 = 10.67 K

22. Mixer (MIX 101)

    Outlet mass flow rate = 125 g/s

23. Heat exchanger 6 (HE VI)

    Pressure drop in hot stream = 0 bar

    Pressure drop in Cold stream = 0 bar

    Minimum Approach ($\theta 6$) = 0.296 K

    Heat Leak = 0.025 KW

    Temperature at 40 = 7.5 K

24. Turbine 3 (T III)

    Efficiency of turbo expander = 64 %

    Outlet pressure = 4 bar

25. Tee (Tee 104)

    Mass flow rate to De Furties = 0.3 g/s

26. Mixer (MIX 102)

    Outlet mass flow rate = 0.7 g/s

27. Heater (H1)

    Outlet temperature = 300 K

    Pressure drop = 0.15 bar

28. Heat exchanger 7 (HE VII)

    Pressure drop in hot stream = 0 bar

    Pressure drop in Cold stream = 0 bar

    Minimum Approach (θ7) = 0.317 K

    Heat Leak = 0.025 KW

    Temperature at 47 = 4.416 K

29. JT Valve

    Pressure drop = 2.8 bar

30. Separator

    Outlet pressure = 1.2 bar

If we change any input value mentioned above all other information is updated automatically in given cyclic configuration.

5.4.1 PROCESS FLOW DIAGRAM OF HELIUM LIQUEFIER IN ASPEN HYSYS:

Figure shows the process flow diagram that drawn in HYSYS. Table shows the steady state properties of all the streams in the PFD, material stream 49 gives the helium liquefaction rate.

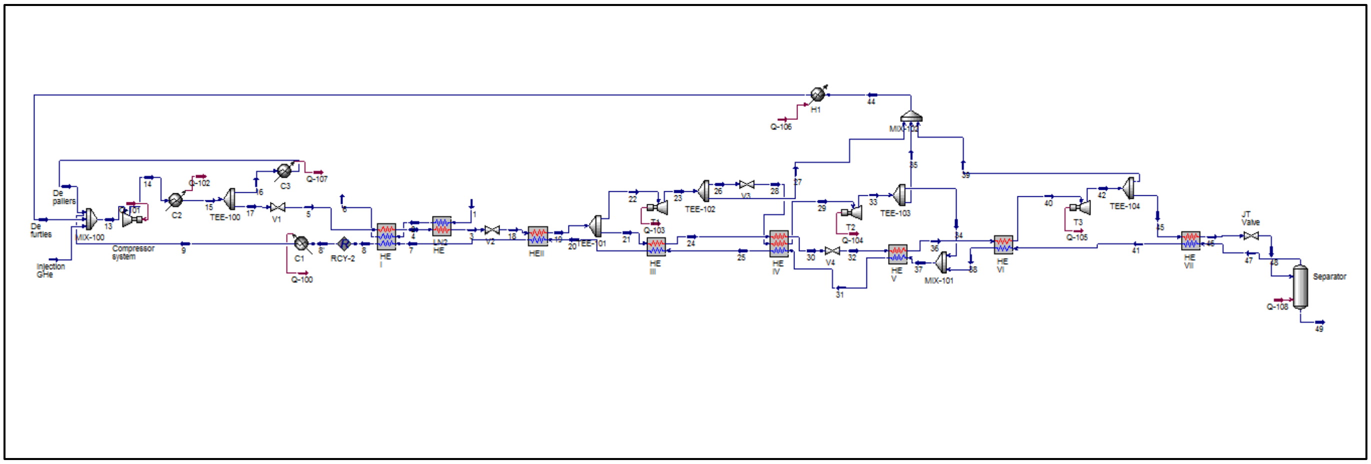


Figure 5.4.1.1: PFD of Helium Liquefier

5.4.2 MATERIAL STREAMS:

Name	1	2	3	4	5	6	7	8	8'	9	De paliers	De furties
Vapour Fraction	0.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Temperature [K]	79.19	79.19	80.00	86.80	310.0	307.0	78.66	307.0	307.0	300.0	300.0	300.0
Pressure [bar]	1.251	1.251	13.20	13.20	13.30	1.251	1.200	1.200	1.200	1.050	1.050	1.050
Molar Flow [kgmole/h]	3.079	3.079	119.4	119.4	119.4	3.079	112.4	112.4	112.4	112.4	7.195	0.6296
Mass Flow [g/s]	23.96	23.96	132.7	132.7	132.7	23.96	125.0	125.0	125.0	125.0	8.000	0.7000
Liquid Volume Flow [m3/h]	0.1649	0.1649	6.392	6.392	6.392	0.1649	6.021	6.021	6.021	6.021	0.3854	3.372e-002
Heat Flow [UserUnit*]	-10.25	-5.534	-149.9	-145.2	8.756	0.2142	-142.4	5.793	5.793	1.243	7.957e-002	6.962e-003
Name	Injection GHe	13	14	15	16	17	18	19	20	21	22	23
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Temperature [K]	300.0	300.0	1027	310.0	310.0	310.0	80.01	35.29	30.43	35.29	35.29	27.32
Pressure [bar]	1.050	1.050	14.00	13.80	13.80	13.80	13.05	12.99	1.200	12.99	12.99	5.400
Molar Flow [kgmole/h]	6.296	126.5	126.5	126.5	7.195	119.4	119.4	119.4	112.4	51.97	67.38	67.38
Mass Flow [g/s]	7.000	140.7	140.7	140.7	8.000	132.7	132.7	132.7	125.0	57.78	74.92	74.92
Liquid Volume Flow [m3/h]	0.3372	6.778	6.778	6.778	0.3854	6.392	6.392	6.392	6.021	2.783	3.609	3.609
Heat Flow [UserUnit*]	6.962e-002	1.399	533.4	9.284	0.5279	8.756	-149.9	-181.2	-173.8	-78.90	-102.3	-105.5
Name	24	25	26	27	28	30	29	31	33	34	35	32
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Temperature [K]	26.92	26.37	27.32	27.32	27.32	15.62	15.62	13.27	10.32	10.32	10.32	15.61
Pressure [bar]	12.99	1.200	5.400	5.400	5.300	12.99	5.300	1.200	1.200	1.200	1.200	12.84
Molar Flow [kgmole/h]	51.97	112.4	67.20	0.1799	67.20	51.97	67.20	112.4	67.20	67.02	0.1799	51.97
Mass Flow [g/s]	57.78	125.0	74.72	0.2000	74.72	57.78	74.72	125.0	74.72	74.52	0.2000	57.78
Liquid Volume Flow [m3/h]	2.783	6.021	3.599	9.634e-003	3.599	2.783	3.599	6.021	3.599	3.590	9.634e-003	2.783
Heat Flow [UserUnit*]	-81.55	-176.5	-105.2	-0.2815	-105.2	-85.35	-110.0	-185.1	-111.8	-111.5	-0.2993	-85.35
Name	36	37	39	44	40	41	38	42	45	46	48	49
Vapour Fraction	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	0.2501	0.0000
Temperature [K]	10.67	10.34	5.982	11.63	7.500	5.660	10.37	5.982	5.982	5.255	4.416	4.416
Pressure [bar]	12.84	1.200	4.000	1.200	12.84	1.200	1.200	4.000	4.000	4.000	1.200	1.200
Molar Flow [kgmole/h]	51.97	112.4	0.2698	0.6296	51.97	45.40	45.40	51.97	51.70	51.70	51.70	6.296
Mass Flow [g/s]	57.78	125.0	0.3000	0.7000	57.78	50.48	50.48	57.78	57.48	57.48	57.48	7.000
Liquid Volume Flow [m3/h]	2.783	6.021	1.445e-002	3.372e-002	2.783	2.432	2.432	2.783	2.769	2.769	2.769	0.3372
Heat Flow [UserUnit*]	-87.30	-187.1	-0.4618	-1.043	-88.65	-76.90	-75.54	-88.95	-88.49	-88.95	-88.95	-10.87
Name	47	** New **										
Vapour Fraction	1.0000											
Temperature [K]	4.416											
Pressure [bar]	1.200											
Molar Flow [kgmole/h]	45.40											
Mass Flow [g/s]	50.48											
Liquid Volume Flow [m3/h]	2.432											
Heat Flow [UserUnit*]	-77.38											

Material Streams | Compositions | Energy Streams | Unit Ops

Table 5.4.2.1: Material streams in Helium Liquefier

5.5 COMPARISION OF ANALYTICAL AND ASPEN HYSYS RESULTS:

Process parameters like temperatures of heat exchanger and turbine, UA of Heat Exchangers calculated using analytical method and Aspen HYSYS results are compared.

- UA values of all Heat Exchangers are tabulated below :
  - ✓ There is hardly 17% variation between UA values calculated analytically for HE IV and 14% for HE VII.
  - ✓ Others are below 10% variation.

	UA Value using Analytical Method (W/C)	UA Value using Aspen Hysys (W/C)
HE I	30037.522	30000
HE II	11448.32773	11930
HE III	1281.415262	1371
HE IV	7734.034239	6393
HE V	2045.627648	2125
HE VI	1612.463348	1491
HE VII	887.6467783	758

Table 5.5.1: Comparison of UA values

- Refrigeration and Liquefaction capacity are tabulated below:
  - ✓ Liquefaction capacity is kept constant and the values of refrigeration capacity calculated using analytical method and by using Aspen are matching with each other.

	UA Value using Analytical Method (W/C)	UA Value using Aspen Hysys (W/C)	UA Values for Existing Plant (W/C)
Liquifaction Capacity (g/s)	7	7	7
Refrigeration Capacity (W)	701.9034928	707.01	650

Table 5.5.2: Comparison of Refrigeration and Liquefaction Capacity

- Turbine inlet outlet temperatures are tabulated below:
  - ✓ Turbine inlet outlet temperatures are almost matching so error is considered to be negligible.

		T I	T II	T III
Using Analytical Method:	Inlet temperatures	35.3	15.62	7.5
	Outlet Temperature	27.32205431	10.31772247	5.981976441
Using Aspen HYSYS:	Inlet temperatures	35.29	15.62	7.5
	Outlet Temperature	27.32	10.32	5.982

Table 5.5.3: Comparison of turbine inlet outlet temperatures

- Heat Exchangers inlet outlet temperatures are tabulated below:
  - ✓ Hot and cold stream inlet outlet temperatures of all Heat Exchangers are matching with each other. Error between those is less than 10%.

			HE I (Cold stream of N2)	HE II	HE III	HE IV (Hot stream of He)	HE V	HE VI	HE VII
Using Analytical Method:	Heat Exchangers hot stream temperatures(K)	Inlet (Thi)	310	80.00714974	35.3	27.32205431	15.60513935	10.67	5.981976441
		Outlet (Tho)	86.7683045	35.3	27.32205431	15.62	10.67	7.5	5.255432267
	Heat Exchangers cold stream temperatures(K)	Inlet (Tci)	78.65714974	30.44353092	26.77205431	13.47358299	10.33908191	5.66197644	4.408
		Outlet (Tco)	307.016	78.65714974	30.44353092	26.77205431	13.47358299	10.3707	5.661976441
	3rd hot or cold stream for 3 stream HE	Inlet (Thi / Tci)	79.19			27.32205431			
		Outlet (Tho / Tco)	307.016			15.62			
Using Aspen HYSYS:	Heat Exchangers hot stream temperatures(K)	Inlet (Thi)	310	80.01	35.29	26.92	15.61	10.67	5.982
		Outlet (Tho)	86.8	35.29	26.92	15.62	10.67	7.5	5.255
	Heat Exchangers cold stream temperatures(K)	Inlet (Tci)	78.66	30.43	26.37	13.27	10.34	5.66	4.416
		Outlet (Tco)	307	78.66	30.43	26.37	13.27	10.37	5.66
	3rd hot or cold stream for 3 stream HE	Inlet (Thi / Tci)	79.19			27.32			
		Outlet (Tho / Tco)	307			15.62			

Table 5.5.4: Comparison of Heat Exchangers hot and cold stream inlet outlet temperatures

All Heat Exchangers hot and cold stream inlet outlet temperatures, Turbines inlet outlet temperatures UA values of all Heat Exchangers are matching with each other with 17% error between analytical calculations and values calculated using Aspen HYSYS. This validates the analytical optimization methodology.

5.6 BEHAVIOR OF HEAT EXCHANGERS IN GIVEN CONFIGURATION:

Different graphs have been plotted for all heat exchangers using Aspen HYSYS. Behavior of all plots for Heat Exchanger I have been discussed below:

I. Plot of Temperature VS Heat Flow for HE I:

- Y axis shows the temperature change along the length of the Heat Exchanger. This plot signifies how the heat is being transferred from hot stream of helium to the cold stream.
- This graph shows the linear variation of properties like Cp.
- For HE VII, Cp values are varying so it does not show a linear relation.

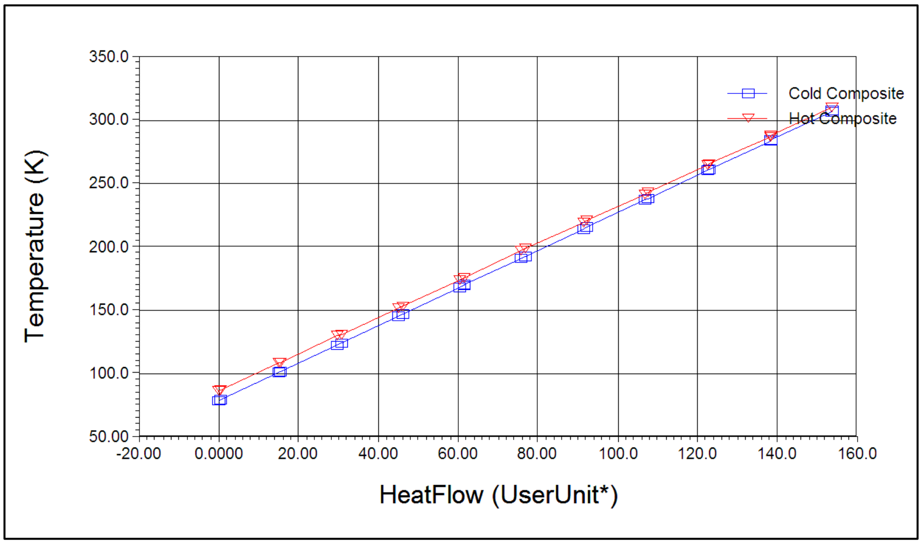


Figure 5.6.1: Plot of Temperature VS Heat Flow for HE I

II. Plot of Temperature VS UA for HE I:

- Along the length of the Heat Exchanger how UA is changing with its Temperature is shown in this plot.
- Gradually delta UA is decreasing as temperature is decreasing because at higher temperatures velocity as well as heat transfer coefficient is high which ultimately increases UA.

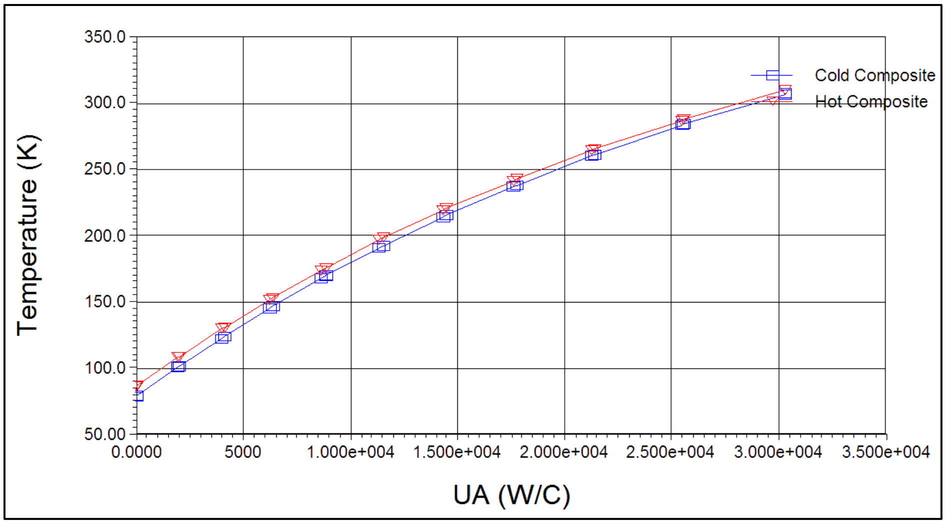


Figure 5.6.2: Plot of Temperature VS UA for HE I

III. Plot of Delta Temperature VS UA for HE I:

- This plot shows how UA is varing as Delta Temperature is changing.
- As Delta T is decreasing Delta UA is increasing.

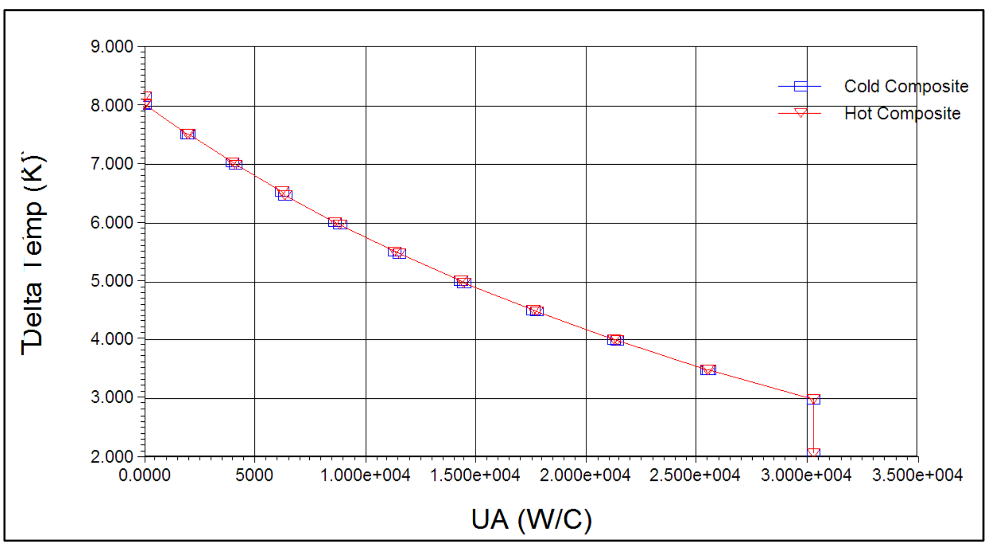


Figure 5.6.3: Plot of Delta Temperature VS UA for HE I

Chapter-6

CONCLUSION

6.1 CONCLUSION:

Among all analyzed methods steady state approach is effectively used for the optimization of process parameters of turbine and heat exchanger such as mass flow rate, temperature, inlet outlet turbine pressures, effectiveness or UA. Using steady state approach analytical method has been developed for 2 compressor system with 3rd turbine and without 3rd turbine for maximizing refrigeration capacity. Same method has been used to analyze the refrigeration capacity and the effect of compressor outlet pressure for 3 compressor system.

Analytically developed procedure for 2 compressor system with 3rd turbine has been validated using Aspen HYSYS. All temperatures and UA values of Heat Exchangers, turbine inlet outlet temperatures calculated using analytical method is found to be matching with the HYSYS results. Effect of compressor outlet pressure is directly proportional to liquid formation at JT outlet and refrigeration capacity with 3rd turbine and without 3rd turbine it increases till certain value then decreases.

6.2 FUTURE WORK:

Process parameters like turbine mass flow rate have to be varied to optimize the refrigeration capacity and liquid formation at JT outlet.

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